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LAND AND MARINE
DIESEL ENGINES.

BY

GIORGIO SUPINO.

TRANSLATED BY

ENG. LIEUT.-COMMDR. A. G. BREMNER, R.N.,

AND

JAMES RICHARDSON, B.SC.(ENG.)LOND., A.M.INST.C.E.

With 19 Plates and 380 Illustrations in the Text.

LONDON:
CHARLES GRIFFIN & COMPANY, LIMITED;
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1915.

TRANSLATORS' PREFACE.

ECONOMIC conditions were more conducive to the development of the oil engine on the Continent of Europe than in the United Kingdom. This was in part due to our advantage in having an abundance of cheap steam-raising coal. As a consequence the theory and practice of the internal combustion engine more fully and readily engaged the attention of Continental engineers and experience in this type of prime mover is more extensive on the Continent of Europe than in this country. The superior thermal efficiency of the oil engine has now, however, won many supporters here, and the fact that it is practically indispensable for certain purposes, notably for submarines, has resulted in a great increase in construction in the United Kingdom. The time has not come, however, for a record based solely on British experience. Indeed, most of the British firms building oil engines have, so far, based their practice on Continental systems. This is particularly the case with marine engines.

No apology is thus needed for the translation into English of a text-book which is widely accepted on the Continent as a standard work, embracing comprehensively, yet without redundancy, existing knowledge of land and marine engines. Ing. Supino, the author, an Italian engineer of high repute, who died ere yet he had had time to enjoy the reputation he had won, made a special study not only of the theory, but of the construction and running of oil engines, and such merits as this book possesses as a translation are due entirely to his engineering genius, erudition, and lucidity of exposition. The translators have sought only to interpret his ideas.

The English Edition, it is hoped, will be accepted as much for its recognition of Ing. Supino's work, as for the stimulus and assistance it may bring to British workers. It is true that the Marine Diesel Engine has not, so far, realised in this country the extensive adoption predicted by optimists. Our success in steam machinery tended to stereotype practice in reciprocating machinery, which had proved thoroughly efficient with steam as the prime mover, and it has not always been

foreseen that the internal combustion engine introduced conditions requiring modification, if not, indeed, entirely new experience. A change is fast disclosing itself, and it is recognised that the oil engine may be made as reliable as, and at the same time more efficient than, the steam engine. Its use for submarines has quickened the interest in its design and manufacture, and Ing. Supino's research in this direction as in merchant ship propulsion, illustrated as it is with many sections, detailed drawings, and formulæ, must be serviceable to the young engineer embarking upon practical work in oil engines, and perhaps, also, of some help to experts as a book of reference recording much of that which has already been done. To facilitate this latter service a comprehensive index, list of illustrations, tables, etc., have been appended.

A word may be said as to the nomenclature, regarding which there is wide divergence in practice. This is inevitable in the case of a new and somewhat complicated product. The translators have attempted consistently to adhere to names for engine parts which, in their experience, have proved most satisfactory. At the same time, they recognise that time and experience may bring about modifications in some cases.

In the original work calculations, dimensions, etc., are given according to the metric system. In this issue British units are added where these tend to a readier understanding of the text. Calculations, however, have not been converted, as these appeal to the more expert reader, who is doubtless conversant with the metric system. A conversion table has been appended.

A section of the book which may be of special value for reference is that dealing with the Rules of Classification Societies for the Construction of Marine Oil Engines, and we wish to express our indebtedness to Lloyd's Registry of British and Foreign Shipping and to The British Corporation for the Classification and Survey of Shipping for their kind permission, enabling us to insert their latest rules for this type of marine engine.

A. G. BREMNER.
J. RICHARDSON.

September, 1915.

AUTHOR'S PREFACE.

In writing the part of this book which deals with land engines I have availed myself of the material of my book, *Motori Diesel* (Hoepli, 1912), the first edition of which is now out of print.

Whilst 4-cycle engines have already passed beyond the critical and experimental stages of their development, those of the 2-cycle type of large power, and marine engines in particular, are in such a state of transition that any discussion regarding them is apt to become almost out of date during production. The chapters dealing with these two latter types treat the subject fully and are entirely new.

For the arrangement of the subject matter, and the fuller explanation of certain points, I have endeavoured to benefit by the observations contained in the reviews of the foreign press, and expressed by such authorities as Colombo, Schröter, Diesel, Saldini, Deschamps, Stodola, Schöttler and Witz, to whom I take this opportunity of tendering my deepest thanks for their kind and flattering criticisms.

I owe also a debt of gratitude to my editors, Comm. Hoepli of Milan, and to Herr Oldenburg of Munich, who produced a German Edition of my first work, well and faithfully translated by my friend Ing. Hans Zeman.

GIORGIO SUPINO.

When the work of supervising the printing of this volume was just finished the author, at the early age of twenty-seven years, died suddenly, 4th April, 1913.

LIST OF PLATES.

CONTENTS.

LIST OF TEXT ILLUSTRATIONS.

LIST OF TABLES.

ABBREVIATIONS OF THE NAMES OF FIRMS CONSTRUCTING DIESEL ENGINES, QUOTED IN THE TEXT AND FIGURES.

ABBREVIATION.	FIRM.
A.B.D.M. Stockholm,	Aktiebolaget Diesels Motorer, Stockholm.
Benz,	Benz & Cie, Rheinische Gasmotoren Fabrik A.G., Mannheim.
Blohm,	Blohm & Voss, Hamburg.
Burmeister,	Burmeister & Wain, Copenhagen.
Carels,	Carels Frères, Ghent.
Cockerill,	Société Anonyme John Cockerill, Seraing.
Dingler,	Dinglersche Maschinenfabrik, A.G., Zweibrücken.
Fiat,	Fiat San Giorgio, Turin.
Frerichs,	J. Frerichs & Co., A.G., Osterholz-Scharmbeck.
G. F. Deutz,	Gasmotoren Fabrik Deutz, Cöln-Deutz.
Germania,	Fried. Krupp, A.G., Germaniawerft, Kiel.
G.M.A.,	A. G. Görlitzer Maschinenbau-Anstalt and Eisengiesserei-Görlitz.
Grazer,	Grazer Waggon und Maschinenfabrik Aktiengesellschaft vorm. Joh. Weitzer-Graz.
Güldner,	Güldner Motoren Gesellschaft, Aschaffenburg.
Harlé,	Harlé & Cie, succ. de Sauter Harlé & Cie, Paris.
Kind,	Ing. Paolo Kind, Turin.
Kolomna,	Société de constructions mécaniques de Kolomna, Kolomna.
Körting,	Gebr. Körting, A.G., Körtingsdorf bei Hanover.
L.W.,	Societa Italiana Langen & Wolf, Milan.
M.A.N.,	Maschinenfabrik Augsburg-Nürnberg, A.G.
Neederlandsche,	Neederlandsche Fabriek van Werktuigenen Spoorweg-Materiel, Amsterdam.
Nobel,	Nobel Frères, Petrograd.
Reiherstieg,	Reiherstieg, Schiffswerfte und Maschinenfabrik, Hamburg.
Richardson,	Richardson, Westgarth & Co., Hartlepool.
Sabathé,	Société des Moteurs Sabathé, La Chaléassière, Saint Étienne.
Savoia,	Cantieri Savoia, Corniglíano Ligure.
S.L.M. Winterthur,	Schweizerische Locomotiv und Maschinenfabrik, Winterthur.
Sulzer,	Gebr. Sulzer, Winterthur.
Tecklenborg,	J. C. Tecklenborg A.G., Bremerhaven, Geestemünde.
Tosi,	Franco Tosi, Legnano.
Weser,	A. G. Weser, Bremen.

LAND AND MARINE DIESEL ENGINES.

PART I.

CHAPTER I.

DIESEL ENGINES FOR STATIONARY PLANTS.

FREQUENTLY the truth is stated that the various systems of prime movers instead of rivalling one another, can be apportioned naturally, regarding suitability of application to the general fields of industrial work. This, however, does not signify that the problem of selection is always one of the most elementary, for, whilst the local conditions often clearly indicate whether, as a general rule, preference should be given to hydraulic, electric or heat engines, it is less easy to decide which system of heat engine lends itself best to the particular case under consideration.

For the greatest powers, the steam turbine, it may be said, has no competitors, but from 2,000 B.H.P. downwards, steam engines, and the various systems of internal combustion engines present, each in turn, advantages which often render indeterminate the problem of selection.

Uncertainties are often due to prejudice or to the personal opinions of the purchaser, but more often to the impossibility of determining fully the working expenses, especially those dealing with repairs, depreciation, etc.

Great weight is usually given to information obtained from existing plants similar to those which it is desired to erect. Such information, although collected from those responsible for the running of any particular plant, is not always absolutely reliable, and in considering the merits of any given type, it may be that the number of such installations actually in service is not sufficient to permit the establishing of reliable mean values for the particulars required.

In certain cases, even where the conditions are particularly suitable for internal combustion engines, there are those who place all their faith in the proverbial safety of the steam engine, and prefer it to the former, perhaps even in spite of greater first cost and running expenses.

In considering internal combustion engines of all types as a whole, it may be stated that, up to the present time, they are regarded in some quarters as being unreliable, and that, whilst some little time ago this opinion may have been entirely justified, its retention at the present time is largely due to the fact that many engineers have not complete knowledge of this new type

of prime mover, and have been unable to follow its rapid progress and to appreciate the consequent specialisation.

For small powers, where only intermittent work is required, petrol engines running at a high speed of revolution are often suitable on account of their low first cost, against which must be set the disadvantages of the high cost of fuel per B.H.P. per hour and their comparatively short life.

For similar powers, engines of the same type running on paraffin are gaining headway on account of their many excellent qualities. The cost for fuel in this case is from 10 to 12 centesimi (1d. to 1·25d.) per B.H.P. per hour, but their first cost is a little higher than that for high-speed petrol engines. Generally, these paraffin engines are preferred if fairly frequent and continuous work is demanded of them.

From 4 or 5 B.H.P. to 12 or 15 B.H.P. heavy oil explosion engines are applicable where the work is fairly continuous and the load fairly steady. For these, the periods of working should be of sufficient duration to counteract the expense and time required to heat the hot bulb with a blow lamp prior to starting.

With these engines the load should be fairly constant, since when running light or at small loads the ignition of the comparatively dense fuel would be incomplete, unless the blow lamp were kept alight. The combustion of the fuel does not develop sufficient heat to maintain the bulb at the requisite temperature. With every variation of the load the amount of the water drip into the cylinder scavenging air—a feature still retained by most makers of hot-bulb engines—must be readjusted. (Chapter V., *et seq.*)

The fuel consumption of this type of engine is considerably less than with paraffin engines, but the conditions enumerated above have the effect of making the competition between these two types exceedingly keen.

Above 15 to 20 B.H.P., the Diesel engine, together with the suction gas engine, enters the competitive field. For such low powers the latter is generally unsuitable, although as a rule more economical in fuel, and lower in first cost as well as in running cost. The high powers attained with multi-cylinder Diesel engines cannot be reached by suction gas engines, except by a multiplicity of units,* and a sacrifice of some of the advantages of economy. Local conditions, of course, may increase or negative the superiority from the economical point of view of either of these prime movers.

For example, where the cost of land is high, and the question of space occupied relatively of great importance, the advantages of the Diesel engine are increased, as it requires only from one-half to two-thirds of the ground area necessary for the installation of a horizontal gas engine (see the table on p. 3).

The 50 to 60 litres (22 to 27 galls.) of water per B.H.P. per hour required by a gas plant would present a difficulty in certain localities in which the supply of the 20 litres (9 galls.) per B.H.P. per hour required by the Diesel engine would be comparatively simple. In comparing the amounts of circulating water required, it should be stated that, with the Diesel engine, all

* At the present time, many Diesel engines of 1,000 to 2,000 B.H.P. are being built, whilst gas engines, as a rule, do not exceed 300 to 400 B.H.P. Large double-acting blast furnace gas engines, owing to the difficulty of constructing generators for the powers reached by these engines, cannot be considered in this comparison of self-contained plants.

COMPARATIVE SPACE REQUIRED FOR SUCTION GAS AND DIESEL ENGINES.

Machinery Space in Feet.	Brake-Horse-Power of Installation.							
	50		100		200		400	
	Gas.[1]	Diesel.[1]	Gas.[1]	Diesel.[1]	Gas.[2]	Diesel.[3]	Gas.[4]	Diesel.[5]
Engine room,	39·4×18·0	19·7×19·7	49·0×19·7	19·7×26·2	49·0×26·2	23·0×26·2	49·0×52·5	24·6×32·8
Gas producer room,	13·1×18·0	..	16·4×19·7	..	18·0×26·2	..	19·7×52·5	..
Total area occupied, sq. ft.,	946	388	1,290	452	1,830	603	3,605	807

[1] Engine of one cylinder.
[2] ,, two cylinders.
[3] ,, three cylinders.
[4] Two engines of two cylinders each.
[5] One engine of four cylinders.

the water may be recovered except the loss from evaporation or leaks at the joints (about 10 per cent.), whilst with the gas generator and scrubber about 16 litres (7 galls.) per B.H.P. per hour is completely lost, and it is thus clearly seen that a scarcity of water may be a decisive argument in favour of the Diesel engine.

There is one class of plant for which the Diesel engine is absolutely suitable as a solution of all the difficulties connected therewith—*i.e.*, those installations in which intermittent running is required, and in which full load must be rapidly attained, such as in reserve sets for electric power stations, drinking water pumping stations, wireless stations, etc. Engines using lighting gas or petrol have, in common with Diesel engines, the important advantage of being able to be started immediately, but the lighting gas engine has the disadvantage of not being self-contained, and the petrol engine for high powers that of the high cost of fuel.

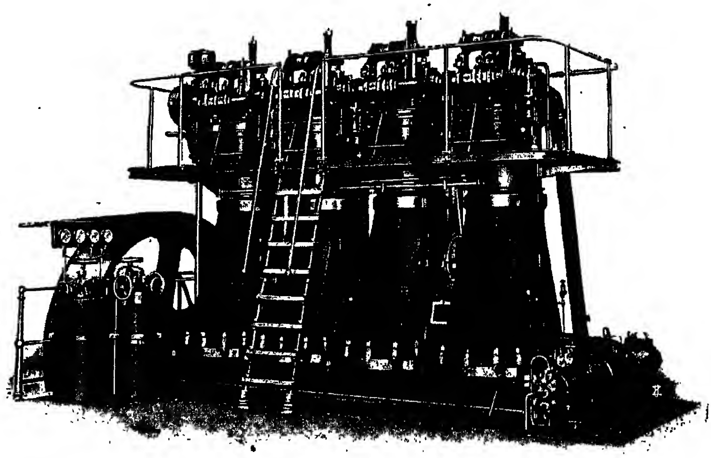

Fig. 1.—Four-Cylinder, Four-Cycle Land Engine.

The suction gas engine is self-contained and is economical, and in general is more suitable than the steam engine for the class of work outlined, but it requires about three hours from the time of lighting up the producer to the time of starting the engine. By keeping the producer always alight (and so considerably increasing the running expenses) the engine may be started after the fan for blowing the fire has been running for 20 or 30 minutes. The effect of the blowing fan is never so great as that of the engine suction, so that the quality of the gas from the producer at the time of starting is not sufficiently good to permit the engine to be put on full load until it has been running for some little time.

When a suction gas engine has run for a period at a considerably reduced load, the producer is not able fully to respond to a sudden increase of load, since the combustion therein cannot be immediately augmented to supply a sufficient quantity of rich gas.

With Diesel engines this disadvantage is not present, for if the load suddenly increases the fuel pumps instantly adapt their delivery thereto, and the engine immediately develops the required output. Should the increase of load be considerable, the pressure of the fuel injection air is raised by opening up the suction of the compressor, or, when a rapid rise is desired, by allowing the compressed air from the reserve storage to pass into the fuel injection air bottle to raise the pressure therein.

In the case of lighting plant for shops, theatres, banks, exhibitions, etc., where power is only required for a few hours per day, the case for the Diesel engine is particularly strong, since when the engine is stopped no fuel is being consumed—*i.e.*, there are no stand-by losses—and where power must be generated in the middle of a city the Diesel engine further presents the advantages of the convenient and rapid storage of fuel, the absence of chimney, of smoke, etc.

For reserve engines, to meet the requirements of compactness and low first cost, high-speed Diesel engines have been constructed (Fig. 2, L.W.), and their main features are described in subsequent chapters.

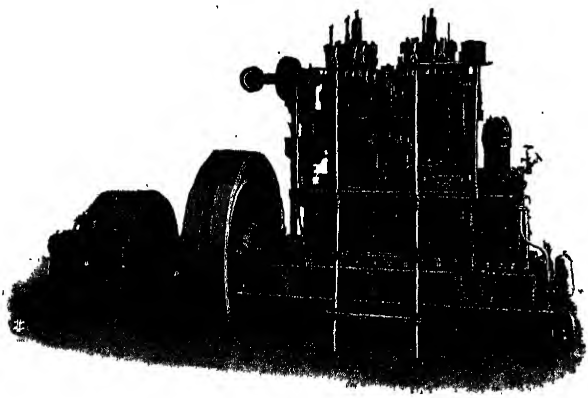

Fig. 2.—Two-Cylinder, Four-Cycle, High-Speed Land Engine.

Although for certain installations the conditions may require such types of high-speed engines, only when the considerations tending towards this application are of the most stringent should such an engine be put forward. The delicate nature of the parts of the high-speed engine in comparison with those of the ordinary land type makes frequent attention necessary, and, further, the maintenance charges are likely to outweigh the saving in first cost, if continuous service be imposed.

With plants for agricultural work, dry dock pumping, electric power stations, reserve sets or sub-stations, lighting sets for fortifications or ships, the high-speed engine is distinctly applicable, due to the small space occupied, the intermittent running required, and the small first cost, if the available personnel is of the requisite high order.

The economy of first cost is not due alone to the engine, but also to the foundations and buildings, and to the smaller overall dimensions of the

machine—dynamo, alternator, centrifugal pump, etc.—driven by the oil engine.

In advertisements and catalogues of makers of Diesel engines the following phrase is often encountered:—"Diesel engines do not require expert attention," which must only be taken in its literal sense, and not be understood to mean that any employee is able to take charge of a Diesel engine. Although regulations may not be imposed, calling for an examination of those in charge of such engines, common sense should counsel the buyer to employ only capable men.

The actual watch-keeping with these engines is of the easiest nature, but even the simplest overhauling and repair and adjustment—part of the usual maintenance routine—must be carefully carried out on account of the delicate and complex nature of the mechanism.

Purchasers should be specifically warned against engineers who always desire to make modifications, since the builders of the engines bring to bear upon their design the fruits of considerable experience, and it is frequently found that modifications suggested by those running the engine cause breakdowns.

When heat is required for warming shops, furnaces or drying rooms, etc., the exhaust gases which leave the engine cylinder at about 400° to 600° C. may be utilised according to the load on the engine. The heat units contained therein represent roughly 28 per cent. of those in the fuel consumed, so that in an engine being supplied with about 2,000 calories (7,950 B.Th.U.) per B.H.P. per hour at full load, 560 calories (2,220 B.Th.U.) are to be found in the exhaust gases; but since these latter are slightly cooled in their passage from the engine to the apparatus for extracting their heat, and leave before they give up all their heat, it may be stated that only 17 per cent. (or about 320 calories—1,300 B.Th.U.) with a temperature of about 150° C.* can be utilised for industrial purposes, either directly or through the medium of steam generated in a suitable boiler.

The water which has circulated through the cylinder jackets, and has served to cool the cylinders, contains about the same number of heat units as the exhaust gas, approximately 27 per cent., or 540 calories (2,140 B.Th.U.) per B.H.P. per hour at a temperature from 60° to 80° C. in small engines and 40° to 50° C. in large engines, and may also be used. For instance, when the exhaust gases are employed for the production of steam in a boiler this water may be used as feed water, giving thus an appreciable increase of power.

In cases where a large quantity of heat is required, or where this heat is called upon to perform important duties, internal combustion engines in general and the Diesel engine in particular are less applicable for the following reasons than steam engines:—

(1) With internal combustion engines the amount of heat in these sources varies with the load on the engine, and is not available when the engine is stopped.

(2) Since internal combustion engines have a higher thermal efficiency than other prime movers, a considerably smaller number of heat units is to be found in the exhaust.

* See also Barth, "Die Wahl einer Betriebes Kraft," *Zeitsch. Ver. deut. Ing.*, No. 41, 1912.

In the case of the steam engine, should the engine be stopped, the boilers may be utilised to produce the requisite heat.

In conclusion, some hints and suggestions dealing with the sending out of enquiries and the comparisons of specifications and tenders may be given.

The information which it is necessary to supply to the makers is as follows :—

(1) Normal and maximum B.H.P.

(2) Number of hours per day that the engine will normally be running.

(3) The types of machines driven by the engine—*e.g.*, dynamos, alternators (whether by belt or direct coupled), pumps, crushing, spinning machines, etc.

(4) The nature of the load, whether it be constant or variable, and in the latter case, if at all possible, between what limits, and whether suddenly or by degrees.

The particulars given under Clauses 3 and 4 make it unnecessary to specify the degree of irregularity of the flywheel, which, in any case, is better left to the maker of the engine.

(5) The height above the sea level at which the engine will work, since increases in altitude cause decreases in power in the following proportion :—

Height above sea level, .	300,	400,	500,	600,	700,	800,	900,	1,000	metres.
	985,	1,310,	1,640,	1,970,	2,300,	2,620,	2,960,	3,280	feet.
Decrease of power, .	2·5,	4·0,	5·5,	7·0,	9·0,	11·0,	13·0,	15·0	per cent.

so that to obtain 100 B.H.P. at a height of 800 metres (2,620 feet) above sea level an engine of 110 B.H.P. is necessary.

The date of delivery is also required, as well as a drawing showing the overall dimensions of the engine, which is always readily furnished makers, in order to permit of the preparation of the rough design of the .. house.

In comparing the tenders, weight should always be given to the reputation and experience of each individual firm in the construction of similar plants, and information, where possible, should be obtained from those responsible for the running of these. Preference should not always be given to the lowest tender; the actual difference of price for equally satisfactory plants cannot be very great, and a certain suspicion may be attached to a comparatively low figure.

From the particulars given in the specification accompanying the tender, those who have had experience can readily recognise the type of engine, and particular attention should be given to the speed of revolution, since Diesel engines of higher speed of revolution ought generally to be lower in price due to their lightness.

The accessories included in the various tenders must be compared with the greatest care, more particularly with reference to the extent of the piping, if included, and as to whether the foundation bolts, exhaust silencers, air reservoirs, are quoted for. The extent of the spare gear that is covered by the price should be noted, as well as the clauses dealing with packing, transport, erection, and instruction in running to those who will be responsible for running the engine after delivery. There are firms in whose tenders certain necessary parts (which must afterwards be bought at extra price) are neglected in order to attract those buyers who, without comparing specifications, limit themselves entirely to the question of price.

The consumption of cooling water generally specified with the tender must not be taken as absolute, since it is generally found necessary in designing the cooling water system to allow for a considerably greater amount.

The fuel consumptions are guaranteed, as a rule, for heavy oil of a calorific value of not less than 10,000 calories per kgm. (18,000 B.Th.U. per lb.), and are stated usually with a tolerance of about 10 per cent.

It is often found in practice, however, that engines constructed by firms of standing satisfy the guarantees as regards consumptions, etc., without the tolerance being taken into account.

The usual figures for guaranteed consumptions per hour at full load are as follows :—

For engines of less than 20 B.H.P. per cylinder, 210 grammes (0·46 lb.) per B.H.P.-hour.

For cylinders up to 40 B.H.P., 200 grammes (0·44 lb.) per B.H.P.-hour.

From 50 to 80 B.H.P., 190 to 195 grammes (0·425 lb.) per B.H.P.-hour, and for powers in excess of 80 to 100 B.H.P. per cylinder, the average may be stated at 185 grammes (0·41 lb.) per B.H.P.-hour.

For three-quarter load the consumption increases by about 5 per cent. and at half-load about 20 per cent. High-speed and two-cycle engines, even of relatively high power, have higher rates of consumption.

In considering the question of consumption of fuel, the density of the fuel should always be taken into account. For instance, in Italy, where fuel oil of a density greater than 0·925 is subject to a reduction of import duty of 2 lire (1s. 8d.) per ton, an engine which can only satisfactorily burn a fuel oil of lower density than this, must be debited with the increased running cost, due to the amount of lighter and generally more expensive oil which has to be added to the heavy oil to reduce its density.

This condition is, of course, particularly applicable to engines designed and constructed in those parts of the world where, due to geographical and other reasons, the most convenient fuel is relatively light.

The maximum guaranteed output of the engine is usually 20 per cent. above the normal. To run an engine at this power for any considerable period of time would be exceedingly dangerous as it involves possibility of breakdown, and the loss consequent upon the engine being out of action would more than counterbalance the advantages of running at an overload for protracted periods. This overload capacity then should only be relied upon to overcome the maximum peaks in the station load curve.

When the first cost of the engine to be installed is known, the overall expenditure may quickly be estimated by including the cost of ground of the building, water service, and carriage of the machinery of the plant to the site. In this estimate, the following percentages are suggestive, and are based on a running time of 3,000 hours per year at an average of three-quarters normal power :—

Interest on capital, ...	5	per cent.	per annum.
Depreciation and maintenance of machinery,	8 to 9	,,	,,
Depreciation and maintenance of buildings,	3 to 4	,,	,,

If the total hours of running be decreased, the diminution in depreci-

ation and maintenance of machinery charges is not in direct proportion, but may be taken as follows:—

For 1,500 hours' running per annum, 7½ to 8 per cent., and for 200 to 300 hours per annum, from 7 to 7½ per cent.

The cost of fuel and the item for wages depend upon local conditions, and are figures which can readily be obtained.

For lubricating oil and waste, the cost of 0·4 centime (0·04d.) per B.H.P. per hour may be taken as representative of general experience.

If water from the ordinary pressure mains be used for the engine, this cost must be added to the running charges.

CHAPTER II.

MARINE DIESEL ENGINES.

THE number of Diesel engines at work at the present time for marine propulsion is large,* but is less, however, than might be supposed, taking into consideration the ships which have been announced as destined to receive such engines; several of these ships, especially those of large tonnage, were laid down with this intention, but were completed with a set of triple-expansion steam engines. On due deliberation, prudent capitalists thought that to be pioneers was not their main function, and that it was preferable to ensure good dividends than to take risks in the interests of progress. As is always the case, the least timid have been the admiralties of the various Powers, who have been the first to apply the heavy oil engine to marine work, commencing on a small scale with 100 to 150 H.P., and increasing to 500 or 600 H.P. In submarines and submersibles, which hitherto were propelled almost exclusively by petrol engines, Diesel engines are now universally installed, owing to their fuel economy and the increased safety consequent upon the type of fuel employed. These naval applications, which from their nature require a high speed of revolution and a light engine, have tended necessarily to the design of special types, rather than directly to the general solution of the marine Diesel engine problem.

Light high-speed engines are always difficult to construct, and are never so reliable as those which are heavy and slow running. Lightness and high speed of revolution are inherent in the principle on which the petrol engine works. This is not the case with the Diesel engine, with the result that this naval application, although having given valuable experience and data for comparison, has not produced decisive results to further the solution of the general problem owing to the special and particularly difficult conditions under which submarine engines work.

It is not many years since the construction of very small or medium tonnage sea-going cargo boats, provided with engines of some hundreds, or at most, a few thousand B.H.P., was inaugurated. These installations, carried out by constructors with great experience of the exigencies peculiar to marine work, have gained immediate success: Diesel engines have been constructed, unprepossessing in appearance—a kind of hybrid between the Diesel and the marine steam engine—which have been, however, of the true marine type.

At the present time there are numerous river tugs, motor launches (Figs. 3 and 4), sailing ships with auxiliary engines (Figs. 5 and 6), and pleasure boats with Diesel engines which have been completely satisfactory.

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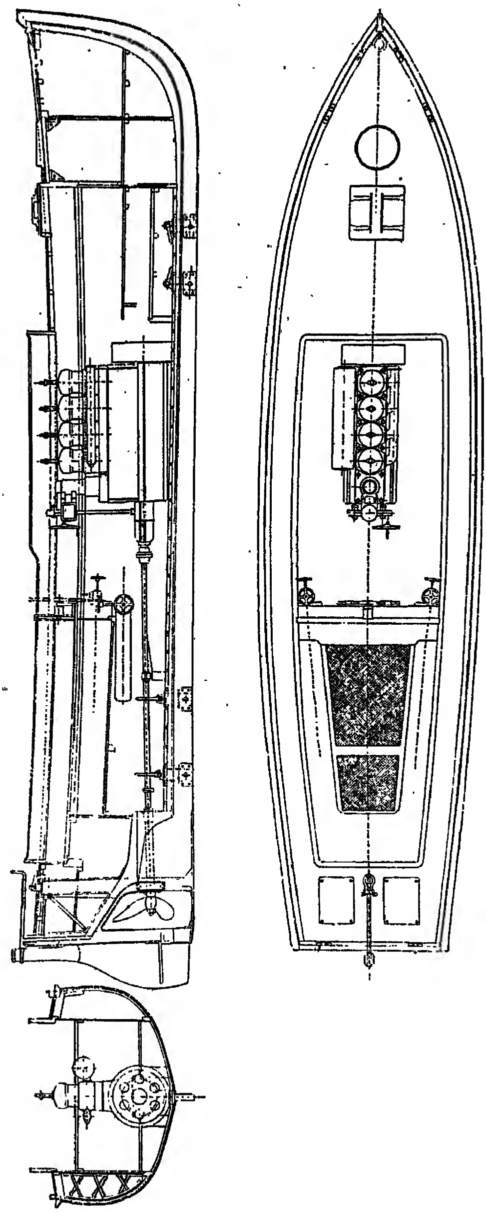
* In November, 1912, the ships in service or in construction propelled by Diesel engines were as follows:—21 tank ships, 37 tugs, 8 sailing ships, 43 passenger and cargo boats, 14 fishing boats, 16 various, 115 submarines, 30 warships (gunboats, torpedo boats, etc.), 20 launches—totalling 304 in all.

The application of heavy oil engines to marine propulsion is making



Figs. 3 and 4.—43-feet Motor Boat, with 100 B.H.P. Diesel Engine (Kind—Turin).

headway, and must increase on the further appreciation of the undoubted merits of the system. It is not to be inferred that all difficulties have been

overcome, and that funnels with their smoke will soon be relegcted to the past. The marine steam engine has such merits of safety, elasticity, and manœuvring capability, has rendered and still renders such good service that, were it for no other reason than the experience and familiarity with which it is regarded, gratitude combined with sentiment would prevent its immediate supercession.

The Diesel engine has not yet been developed for the highest outputs (70,000 to 80,000 B.H.P.), easily obtainable with the steam turbine, and is

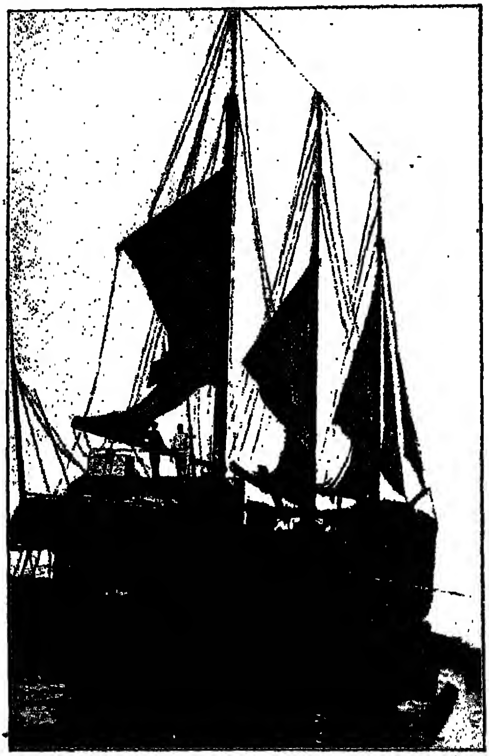


Fig. 5.—Three-masted schooner "Aquila," of 500 tons burthen, with Auxiliary Diesel Engine of 150 B.H.P. (Sulzer).

at present rather complicated and uncertain for small powers. A great future may confidently be anticipated for this new type of engine, but the service still to be rendered by the world's coal deposits cannot be denied.

The advantages which practice has shown the Diesel engine to offer for marine propulsion may now be examined. The most important of these is that of fuel economy—a great advantage in all cases—though its importance depends to some extent upon the service for which the ship is intended. Whilst steam engines require little less than 1 kilo. (2·2 lbs.) of coal per B.H.P.

per hour, a Diesel engine (provided it be not of too small power) consumes about 200 grammes (0·44 lb.) per B.H.P. per hour of heavy oil of 10,000 calories per kilo. (18,000 B.Th.U. per lb.). Putting the price of coal delivered at Genoa at about 30 lire (24s.) a ton, 1 B.H.P.-hour with steam costs 3·0 centimes (0·3d.), whilst with the Diesel engine the cost of 1 B.H.P.-hour is about 1·5 centimes (0·15d.), assuming that the price of fuel oil is about 70 to 80 lire (56s. to 64s.) per ton.

As already stated, the advantage of economical fuel consumption has a relative value according to the type or service of the ship. This applies not only to the three broad divisions of war, mercantile, or pleasure vessels, but also to the significance assumed by the saving of fuel effected during "stand-bys," since with the Diesel engine the consumption of fuel is limited to actual running time.

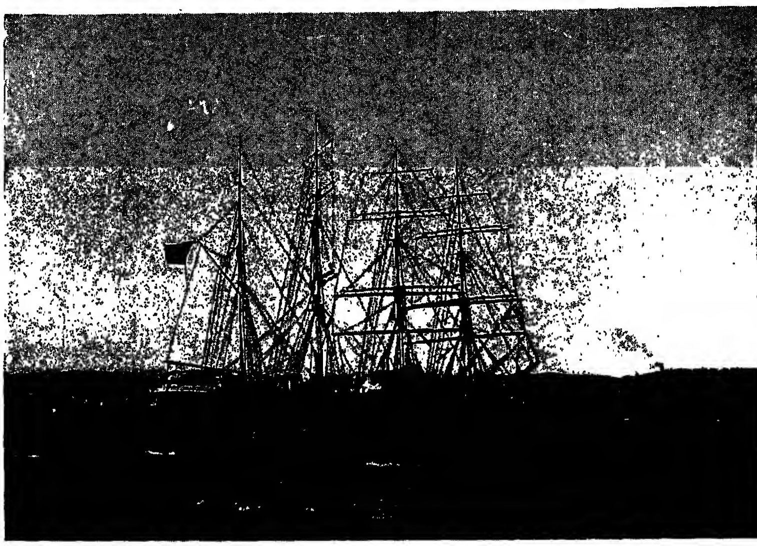
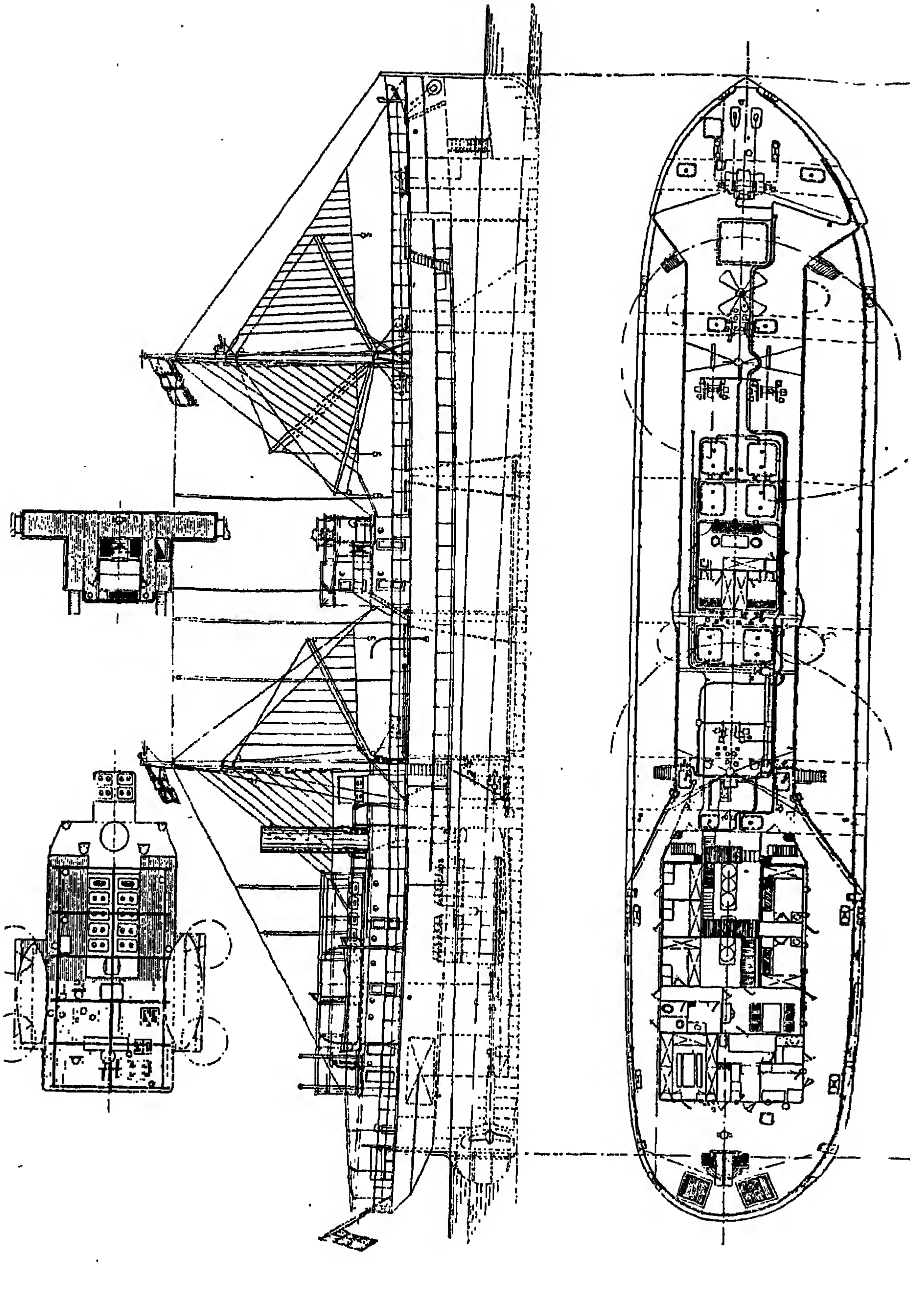


Fig. 6.—Four-masted barque "Quevilly" (France), of 6,000 tons, with Two Auxiliary Diesel Engines, each of 300 B.H.P. (M.A.N. Nürnberg).

This advantage is obviously of little importance for ships making long non-stop voyages, but is considerable in the case of coasting vessels, and those which generally have frequent and lengthy periods at rest, when the boilers have to be maintained under pressure.

Rapidity of getting under way is an asset to warships, pilot vessels, salvage ships, tugs, and for sailing ships with auxiliary engines.

The personnel necessary for manœuvring and watch-keeping in the engineering department is appreciably reduced. The number of such in the engine room for equal powers is slightly less in the case of the steam engine, but the numerous firemen and trimmers are completely eliminated with oil engine installations. For example, the "Selandia," with 2,000 B.H.P., has, for the main engines and all the auxiliary machinery, four engineers (of whom



Figs. 7 to 10.—Tank ship "Vulcanus," of 1,900 tons displacement, with a Diesel Engine of 500 B.H.P. (Werkspoor).

one serves also as electrician), four assistants, and two greasers, whereas with steam engines three engineers, three assistants, and ten or twelve firemen, greasers, and trimmers would have been required.

A motor ship of 7,700 tons, constructed by the Germaniawerft of Kiel for the Deutsch-Americanische Petroleum Gesellschaft, with engines of 2,300 B.H.P., has an engineering personnel of ten, whilst with steam engines she would have needed eighteen.

The Dutch motor tank ship "Vulcanus" (Figs. 7 to 10) has only four men for her engines of 500 B.H.P., whereas if it had a steam engine instead there would be at least six.

In the cargo ship "Monte Penedo" (Figs. 11 and 12), of 6,500 tons, with two Sulzer engines of 1,600 B.H.P. total, the employment of Diesel engines has effected a saving of ten men.

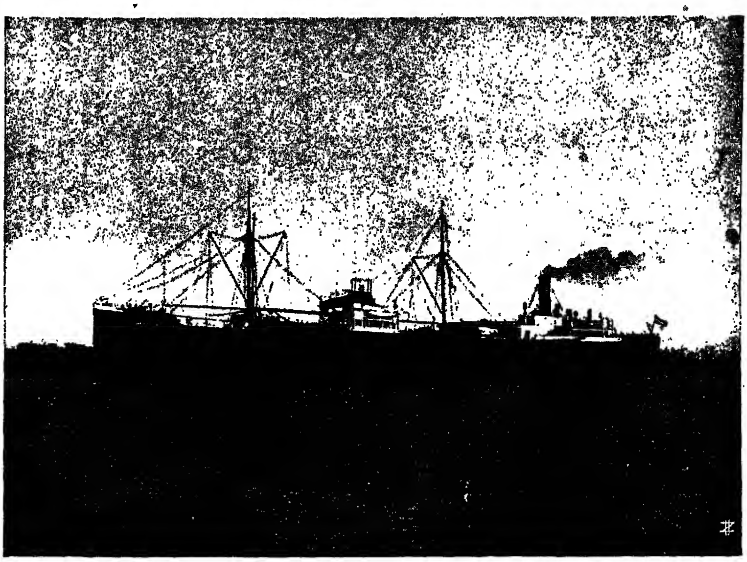


Fig. 11.—M.V. "Monte Penedo."

The personnel of the engine-room department works under less arduous conditions, because the labour of stoking and the heavy work of coal trimming are saved, and the engine itself radiates less heat than a steam engine.

Moreover, the suction of the air for combustion is frequently from the engine room itself, which contributes to efficient ventilation.

By the application of the heavy oil engine economy of space results, but in general the whole of the volume thus gained is not assigned to cargo, part being used to make the engine room more spacious. On this important point of the economy of space it is desirable to be exact. It might be thought that, when substituting a Diesel engine in a ship previously fitted with a steam engine, the new engine might be installed in the place of the old one, and the space previously occupied by the boilers

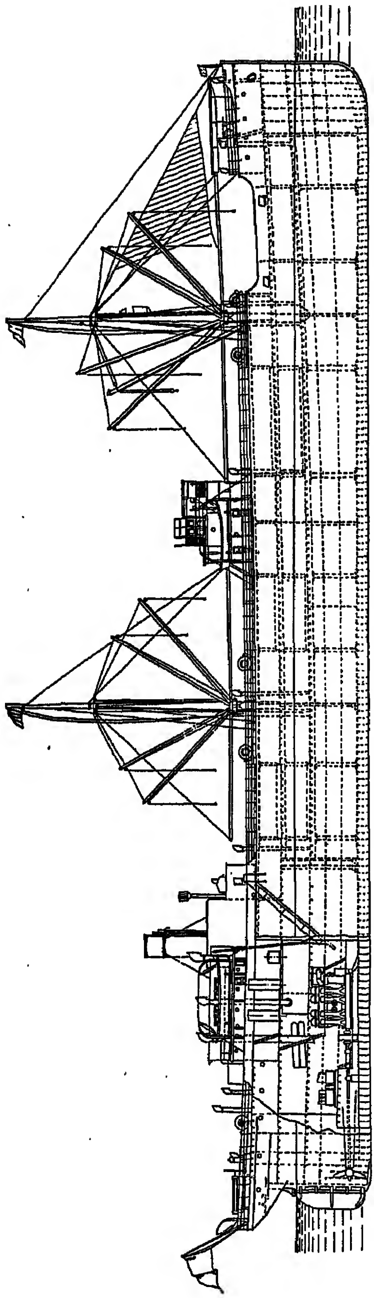


Fig. 12.—M.V. "Monte Penedo," of 6,500 tons burthen, with Two Diesel Engines, each of 800 B.H.P. (Sulzer).

given over to cargo. This is untrue if an exact comparison be made, taking equal speeds of revolutions. From substitutions already made and from simple outline drawings of the machinery, it is seen that the Diesel engine is, on its own account, so much larger than the steam engine of equal power and equal speed of revolution, and the necessary auxiliary machinery to the Diesel engine is so important that the whole plant occupies little less space than that required by the equivalent steam plant with boilers included.

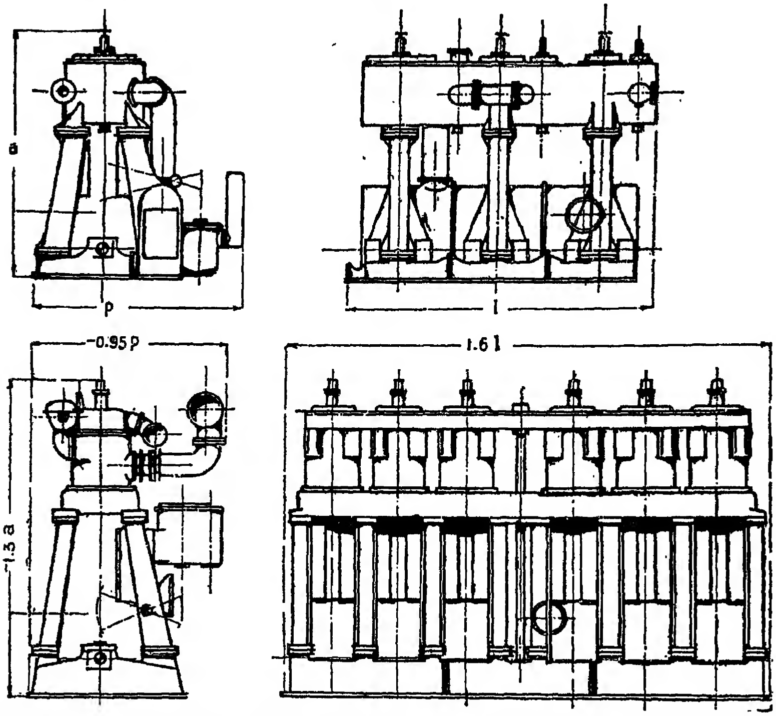
Figs. 13 and 14 represent to the same scale a six-cylinder two-stroke engine of the Germaniawerft of Kiel, giving 1,150 B.H.P. at 140 revolutions per minute, and a three-cylinder triple-expansion engine made by Messrs. Odero & Co.

In Figs. 15 and 16 one of the two four-cylinder two-stroke engines of 800 B.H.P. at 160 revolutions per minute, constructed by Gebrüder Sulzer for the motor ship "Monte Penedo," is compared with an equivalent steam engine built by Messrs. Pattison, of Naples.

Figs. 17 to 25 show the alteration carried out by the Reiherstieg yard of Hamburg in the case of the tank ship "Excelsior," on board of which a triple-expansion engine supplied with steam by two return-tube double-ended marine boilers was replaced by a two-cycle, single-acting, six-cylinder Diesel engine.

From these Figs. 17 to 25, it may be seen that the Diesel plant, with its auxiliaries, occupies practically the same amount of space as the engines and boilers of the original installation—that is to

say, there was no economy of space. A considerable saving, however, is shown in Fig. 26, which represents an alteration carried out by Messrs Sulzer in the case of the steamship " Uto," in service on Lake Zurich, of 62 tons [length 30·3 metres (100 feet), breadth 3·96 metres (13 feet), draught 1·85 metres (6 feet)], in which a four-cylinder, two-cycle Diesel engine, of 150 B.H.P. at 300 revolutions per minute, was substituted for a steam engine of 70 B.H.P. In this way the original engine and boiler space was shortened by 1·3 metres (4·25 feet), and although containing almost twice the power, it was considerably more roomy. It should be noted that in this particular case the propeller speed was considerably increased.



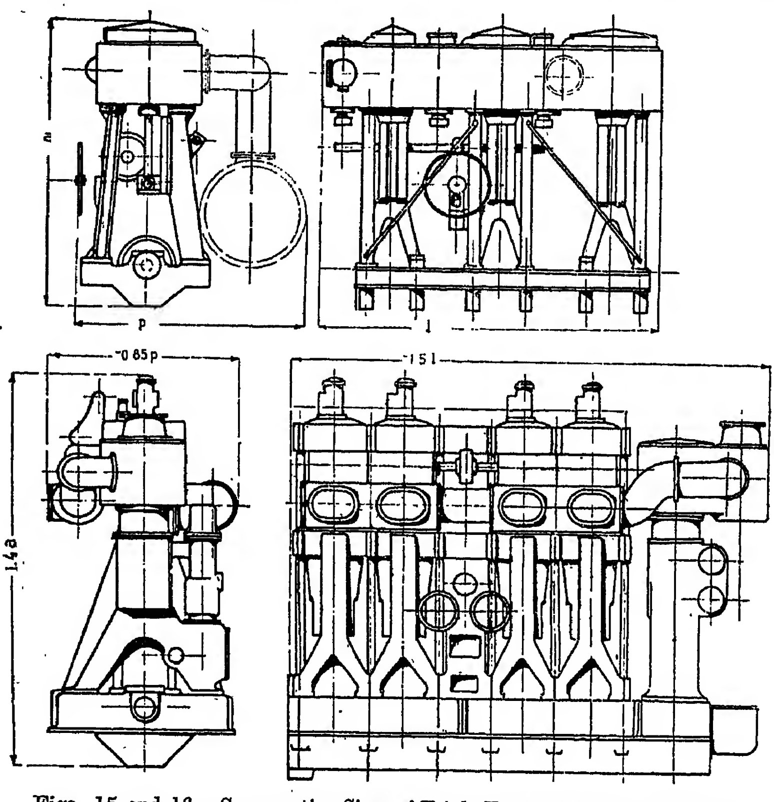
Figs. 13 and 14.—Comparative Sizes of Triple-Expansion Steam Engine and Six-Cylinder Two-Cycle Diesel Engine, each of 1,150 B.H.P.

Again, the Italian motor ship, " Romagna," shipwrecked in November, 1911, in the Adriatic Sea, was provided with two Diesel engines, each of 400 B.H.P., and the engine-room was small (see Figs. 27 and 28). These engines at full power ran at 225 revolutions per minute, whereas a steam engine of equal power suitable for this type of ship would seldom be designed with a speed higher than 160 to 180 revolutions.

If then a comparison be made between the spaces occupied by the two types of engines (in each case the revolutions being the same), and account be taken of their auxiliaries, but not of the fuel stowage, the economy of space which is realised with a Diesel engine is not great. It is well to

add that too great insistence upon the basis of equal speeds of revolution may in reality affect the accuracy of comparisons, since the number of revolutions most suitable for a Diesel engine, in which no particular condition influences its speed, is always higher than that customary for the normal speed of a steam engine of equal power.

An indisputable saving of space is obtained with a Diesel plant, due to the smaller stowage of fuel necessary to give the ship a certain radius of action. Assuming the specific gravity of heavy oil (about 0·9) and of broken coal (0·8 to 1·0) to be practically equal, then the volume and the weight

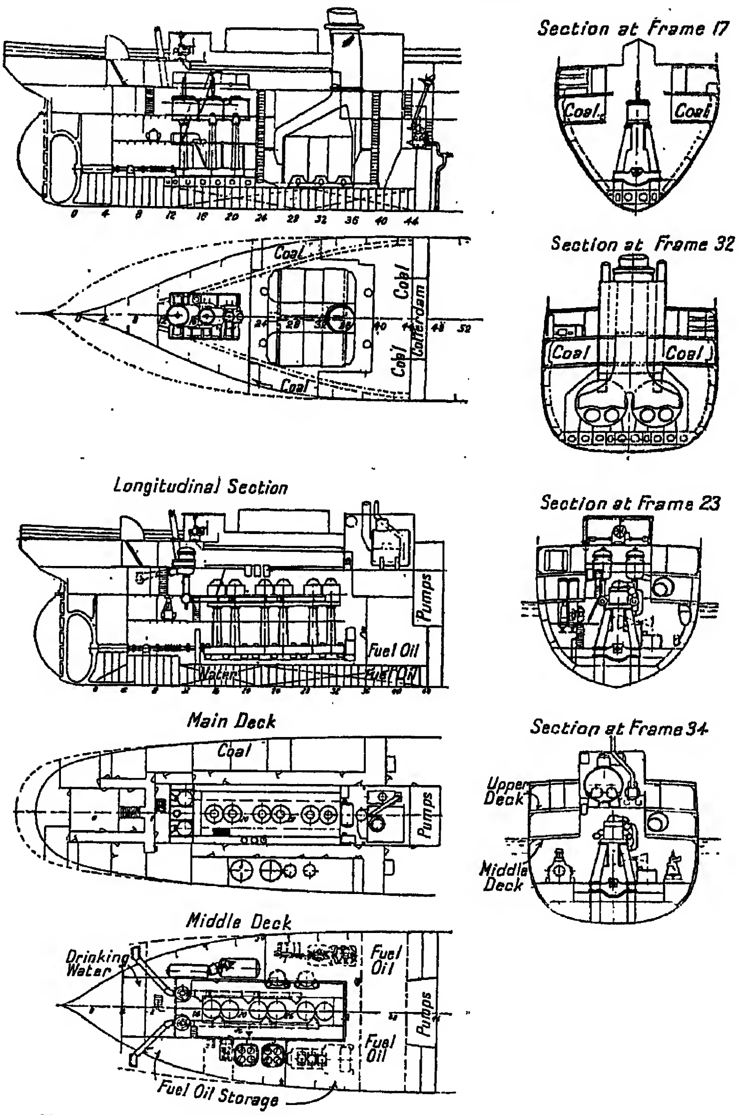


Figs. 15 and 16.—Comparative Sizes of Triple-Expansion Steam Engine and Four-Cylinder Two-Cycle Diesel Engine, each of 800 B.H.P.

of fuel for equal powers and equal radii of action will be reduced by four-fifths, since a Diesel engine consumes less than 200 grammes (0·44 lb.), whilst a steam engine requires about 1 kilo. (2·2 lbs.) of coal per B.H.P. per hour. Further, whilst coal requires ample and accessible stowage, oil may be stowed in the double bottoms of the ship, to be delivered by small pumps to a small reservoir in the engine room.

The taking in of the fuel becomes, in addition, more easy, rapid, and clean, as it may be carried out directly from shore reservoirs or tank waggons by

means of pumps. In the case of warships, rapid replenishment, even at sea, by means of flexible hoses is possible, dispensing thus with the necessity of the supply ship coming close alongside the vessel to be supplied, a proceeding often dangerous and even impossible in heavy weather.



Figs. 17 to 25.—Steam Machinery replaced by Diesel Engines.

For equal radius of action, the economy of weight as of space, due to fuel carried, is about four-fifths, and this figure is confirmed by experience;

the "Robert Nobel," of the firm Nobel, of Petrograd, consumed 48,900 kgs. (48 tons) of coal on the voyage from Baku to Astrakan, whereas, after being fitted with Diesel engines, only 9,780 kgs. (9·6 tons) of heavy oil were used.

The economy in weight of the machinery is much less appreciable than that of the fuel carried, but is far from negligible.

For example, the engine of the motor ship "Vulcanus," of 500 B.H.P. at 180 revolutions per minute, weighs 42 tons—*i.e.*, 84 kgs. (185 lbs.) per B.H.P., and the complete plant, with piping, reservoirs, etc., 85 tons, equal to 168 kgs. (370 lbs.) per B.H.P. The corresponding steam installation, allowing about 60 kgs. (132 lbs.) per B.H.P. for the engine, 110 kgs. (242 lbs.) for the boilers, water, funnel, etc., and 30 kgs. (66 lbs.) per B.H.P. of main engines for the auxiliary machinery and piping, would weigh altogether about 200 kgs. (440 lbs.) per B.H.P.

The complete machinery of the "Monte Penedo," giving 1,600 B.H.P., weighs 160 tons (the two engines weigh about 110 tons, the piping, reservoirs, and accessories 44 tons, the reserve air compressor, complete, about 6 tons)—

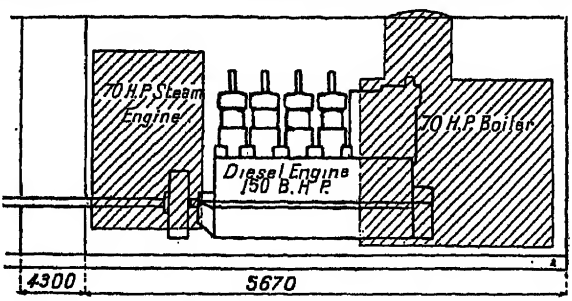


Fig. 26.—Saving in Space effected by substituting High-speed Diesel Engine for Steam Machinery.

that is to say, little more than 100 kgs. (220 lbs.) per B.H.P., whilst if the ship had been provided with a steam installation the weight of this would have been about 180 to 200 kgs. (396 to 440 lbs.) per B.H.P.

The light type of engine for torpedo boats and submarines, including accessories, but excluding auxiliary machinery, weighs from 15 to 22 kgs. (33 to 48·5 lbs.) per B.H.P., whilst steam engines for the same are rarely less than 20 to 25 kgs. (44 to 55 lbs.) per B.H.P.

For example, the six-cylinder Sulzer engine of 300 to 330 B.H.P. for submersibles weighs :—

Engine alone,	4,270 kgs. =	9,400 lbs.
Starting air reservoirs (200 litres),	350 ,,	770 ,,
Injection air reservoirs (40 litres),	70 ,,	154 ,,
Valves and air piping,	60 ,,	132 ,,
Oil coolers,	75 ,,	165 ,,
Oil filters and piping,	20 ,,	44 ,,
Cooling water in engine, etc.,	155 ,,	342 ,,
Total,	5,000 ,,	11,007 ,,

equal to about 16·5 kgs. (36 lbs.) per B.H.P.

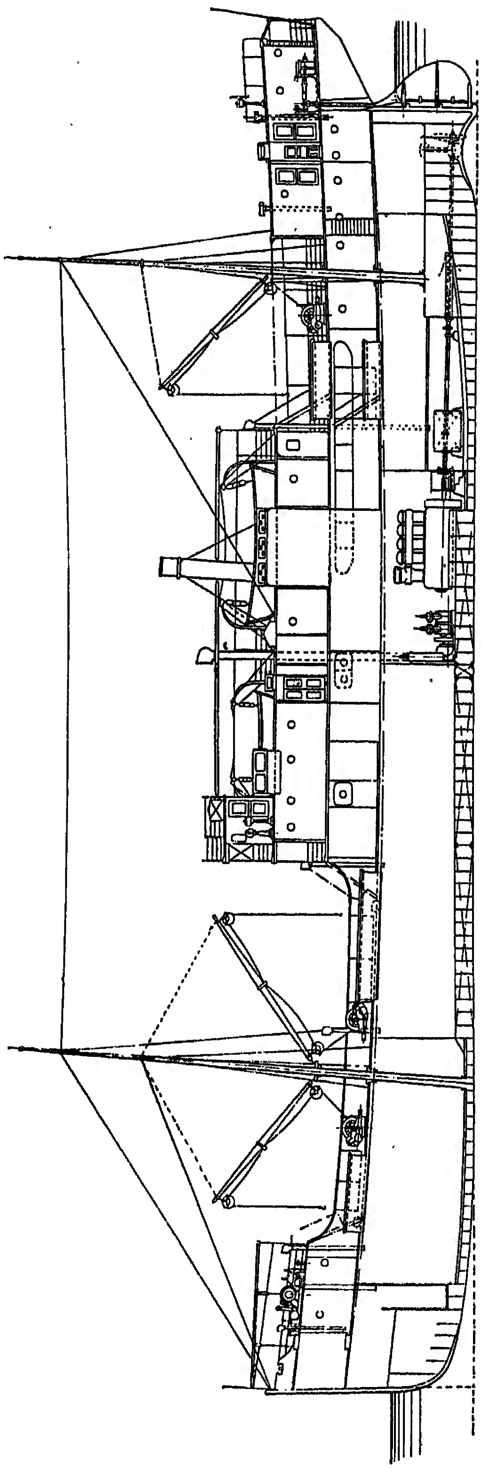


Fig. 27.—Cargo and Passenger Ship "Romagna," 1,000 tons displacement—Two Diesel Engines, each of 400 B.H.P. (Sulzer).

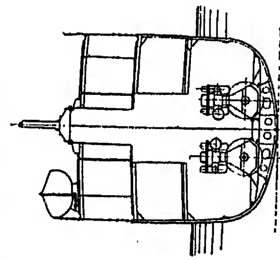


Fig. 28.

The M.A.N. engine of the lightest type of 850 B.H.P. at 450 revolutions weighs with accessories 13,000 kgs. (28,600 lbs.) — *i.e.*, a little more than 15 kgs. (33 lbs.) per B.H.P.

Diesel installations of any considerable power, however, require special auxiliaries, important amongst which for weight, volume, and cost may be reckoned the reserve air compressor, or compressors, with their own engines.

When heavy oil engines were first applied to marine propulsion, a great point was made of the advantages accruing from the elimination of smoke and of the funnels. With the engines running well at about normal load the exhaust is practically colourless, sometimes of a faint bluish tinge. The products of combustion are large in volume and are unbreathable, so that suitable provision must be made for their disposal, to prevent them from affecting the crew, or, when the ship is under way in harbours, from

from inconveniencing those working on the quays, whatever the direction of the ship or of the wind.

The advantage of invisibility is considerable in the case of war vessels, as the ordinary funnels serve as an excellent target for the enemy, produce disastrous results should they be shot away, and reduce the arc of training of the large guns. With sailing ships, also, a funnel would interfere with the rigging and would reduce the available sail area.

In ordinary cargo boats, where invisibility does not count, the most suitable method of disposing of the exhaust gases is, so far, by means of the funnel, which usually has dimensions similar to those required by boilers of the same power, and is frequently used with oil engines for accommodating a silencer or as an aid to engine-room ventilation.

Carrying the exhaust overboard under the counter, as is the practice with motor boats, presents the difficulty that, with an astern wind of greater speed than the ship, the vessel is enveloped by the exhaust gases; further, should the engine position be about amidships, it is undesirable to carry aft, through almost half the length of the ship, the large, hot and noisy exhaust pipes. Under-water discharge of the exhaust gases would go far to eliminate most of the difficulties, but sufficient reliance cannot be placed on a non-return valve for precluding the possibility of water being drawn back through the exhaust pipe into the engine cylinders, with possible disastrous consequences.

Since compressed air is required in large quantities for the fuel injection and starting of the main engines, it is frequently used in motor ships for the duties generally assigned to steam. The syren and steering engine most frequently are worked by this medium, and sometimes also the cargo winches and the capstan engine, which are cheap and of the same design as for steam. Compressed air is expensive, and electric winches have so far made little headway in general application in the merchant service, so that the only remaining solution, that of fitting an oil-fired donkey boiler, is generally favoured in present-day motor ships. This boiler serves for ship heating and for driving the auxiliaries, such as the compressor, dynamo, etc. A second compressor and dynamo are sometimes fitted, each driven by its own Diesel engine, and the cooling water of the oil engines, or water heated by the exhaust gases, is used to facilitate quick generation of steam upon lighting up the donkey boiler.

The fact that the cargo winches and those auxiliaries where absolute reliability is indispensable, such as the compressor and the stand-by dynamo set, are steam-driven, might appear to detract seriously from the credit of the oil engine, and perhaps to some extent does so. The degree of specialisation characteristic of modern engineering practice must be considered, and whereas the Diesel engine is superior to the steam engine from many points of view for the main propelling machinery, it cannot be said to lend itself to the rough and ready duty required of the roughly handled cargo winch.

From the point of view of reliability, it is extremely difficult for any other type of prime mover advantageously to compete with the steam engine, as undoubtedly the latter does not readily break down completely, and can generally be kept running although seriously out of adjustment. On the other hand, the slightest derangement of any of the important and delicate

mechanisms of the internal combustion engine often involves entire stoppage.

In conclusion, it may be said that, to-day, the extended application of the Diesel engine to marine propulsion can be confidently anticipated in the near future. The general problem has been solved, the details have been tried in many ways, and only to experience can be left decisions regardir~ the relative merits of the various systems.

CHAPTER III.

FUELS FOR DIESEL ENGINES—HEAVY OIL EXPLOSION ENGINES AND CONSTANT-PRESSURE OR DIESEL ENGINES.

Diesel engines are capable of burning almost all known liquid fuels, provided they are sufficiently free from sand, earthy matter and sulphur.* Naturally only a limited number of these fuels are employed in practice, those which have been chosen being the most economical and the most suitable for the purpose. Heavy oil, derived from natural petroleum by fractional distillation, is by far the most widely used fuel for Diesel engines. It is exclusively used in the countries rich in petroleum deposits, and large quantities of it are imported where it is not a natural product.

In those countries where the Customs duties for petroleum are heavy, other oils obtained more cheaply as by-products in many manufactures may be and are advantageously employed; for example, the oils from tar, lignite, coal and shale; and the vegetable oils of rapeseed, castor oil, etc.

Heavy oil is a liquid mixture of hydrocarbons rich in carbon, viscous, brown in colour, of penetrating odour and very slightly transparent. Its density lies between 0·85 and 0·95, and its calorific value is generally not less than 10,000 nor greater than 11,800 calories (18,000 to 21,000 B.Th.U. per lb.). As already stated, it is obtained by fractional distillation of petroleum, evaporating at over 300° C., after the lighting and before the lubricating oils are given off. Distillation is naturally carried out in those countries where petroleum is found, and from these the heavy oil is procured. The world's sources of this product are (1) the United States of America, especially California, the Central States, and Illinois; (2) Russia, whose most important wells are at Baku on the Caspian Sea; (3) Roumania (Prahova, etc.); (4) Galicia (Tustanowice, Boryslaw); (5) the Dutch Indies (Borneo, Java and Sumatra). The whole world's production of petroleum, which in 1910 reached 43 million tons, was divided as follows :—

United States,	63·89 per cent.
Russia,	21·48 ,,
Galicia,	3·79 ,,
Dutch Indies,	3·37 ,,
Roumania,	2·97 ,,
India,	1·87 ,,
Mexico,	1·02 ,,
Japan,	0·59 ,,
Peru,	0·40 ,,
Germany,	0·32 ,,
Italy,	0·12 ,,
Canada,	0·10 ,,
Other Countries,	0·08 ,,

* The sulphur should preferably be kept below 1 per cent.

Since the products of the various oil-producing areas differ in composition, the percentage of heavy oil obtainable in each country is not in proportion to the petroleum, but is always between 8 and 13 per cent. of the latter.

Of late years, the adoption of heavy oil for firing steam boilers has gained ground.

In the districts near the sources of production, the use of natural petroleum is even more convenient, since its price is somewhere between 25 and 30 lire (20s. to 24s.) per ton. In those countries of Central and Western Europe, which import heavy oil, the price per ton reaches 50 to 70 lire (40s. to 56s.), excluding import duties which, in some countries, such as Germany, France and Spain, are heavy.

Commercial treaties between the various countries, even more than distance, affect the importation of heavy oil from one or other of the different sources of supply. In Italy, Roumanian oil is most used, because, whilst that from other sources is taxed at the rate of 8 lire per 100 kgs. (6s. 6d. per 2,205 lbs.), a special agreement with Roumania concedes the preferential tariff of 0·2 lire per 100 kgs., provided it satisfies the following conditions :—

Specific gravity above 0·925 at 15° C.
Temperature of distillation above 310° C.
Flash point between 120° and 140° C.
Foreign matter below 20 per cent.

The price of this oil is about 60 to 70 lire (48s. to 56s.) per ton in Italy.

In France, Germany, and Spain,* where the import duty on heavy oil is still very high, fuels of home production or free of duty are used for Diesel engines. Amongst these, during the last few years, tar oil, derived from the distillation of the tars of lignite or coal, has assumed a position of particular importance.

Lignite tar oil (paraffin oil) is a good fuel, but is not cheap ; coal-tar oil is comparatively cheap,† but has too high a flash point to be used alone. After a series of trilas with mixtures of the two oils, latterly, and especially in Germany, the adoption of coal-tar oil as the principal fuel, aided at the moment of ignition by a small quantity of petroleum oil, has become common. The latter is injected into the cylinder first, ignites, and so produces a temperature capable of completely burning the tar oil. From careful trials it is found that the igniting fuel necessary for reliable running of the engine is 2 per cent. at full load, 7½ per cent. at three-quarters, and 13 per cent. at half-load. Since tar oil has a calorific value slightly less than heavy oil, the fuel consumption of the engine increases correspondingly by 12 to 15 per cent. by weight.

Heavy Oil Engines are classed as internal combustion engines, as they convert the heat energy of the fuel into work in the engine cylinder itself. The heavy oil, injected into the cylinder in a suitable condition, ignites, burning with the oxygen of the air therein, and so evolves heat.

In the case of all liquid fuel engines it is necessary, to facilitate and acceler-

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* The duty in Germany is 36 marks (36s.) per ton, in France 90 francs (72s.), and Spain 250 pesetas (100s.).

† In Germany it costs about 30 marks (30s.) per ton.

ate combustion, that at the moment of ignition the fuel should be already intimately mixed with the air, either in the form of vapour or divided into minute particles as spray.

With volatile fuels, such as petrol, vaporisation does not present any difficulty, as it is sufficient to cause the air to bubble through the fuel on its way to the cylinder, or to inject a jet of fuel into the current of air in the inlet pipe. Thus, even without previously heating either the petrol or the air, the latter becomes charged with petrol vapour or carburetted.

In the same way carburation may be effected with alcohol, benzine, and even with paraffin, provided that the temperature is kept high in the case of paraffin.

With heavy oils vaporisation may also be obtained, but only with a very high temperature, and it is generally more suitable to have recourse to mechanical spraying.

Both of these systems of splitting up the fuel are utilised, and lead to two types of engines, different both from the mechanical and the thermodynamic point of view.

The most common method for obtaining vaporisation of the fuel is that of spraying it against a very hot metallic wall of a chamber in communication with the cylinder. On contact with this wall the fuel is heated and vaporised, and since the vaporisation is effected almost instantaneously, the combustion assumes the character of an explosion. Engines working on this principle are known as **heavy oil explosion engines.**

On the other hand, the mechanical means most frequently adopted for obtaining pulverisation or spraying is that of injecting the heavy oil into the cylinder by means of a current of air at a pressure considerably higher than that present in the cylinder itself; thus the fuel is subdivided into minute particles and forms a kind of mist. If the air in the cylinder at the instant when the injection takes place is at a sufficiently high temperature, this mist of oil spontaneously ignites, and the combustion lasts the whole time during which the oil continues to enter, assuming the character of gradual combustion as opposed to an explosion. In this way the combustion takes place in **constant-pressure, or Diesel, heavy oil engines.**

The different method of igniting the fuel leads to an altered thermodynamic action during the stroke in which it occurs, and so to an important modification of the thermal cycle of the engine.

Heavy oil engines work on the four- or the two-stroke cycle principle. In the case of four-cycle engines, as is known, during the first stroke, the piston moving away from the cylinder head draws in air from the atmosphere through the suction valve; during the second stroke all communication with the exterior is closed, and the piston returning towards the cylinder head, compresses the air and so raises its temperature; when compression is complete the fuel oil is injected, ignites, and generates heat, which during the third stroke is transformed into the mechanical work produced by the expansion of the burning gases. During the fourth and last stroke communication is again opened with the exterior, and the piston, on its inward stroke, drives out the burnt gases, leaving the cylinder ready to recommence the cycle.

With the two-stroke cycle, however, all the operations occur in two strokes—*i.e.*, in a single revolution of the crank. The cycle starts from

the position where the piston is closest to the cylinder head, the compression stroke having just been completed, and at the moment fuel enters and is ignited. At this point the first or power stroke commences—*i.e.*, the expansion stroke. When the piston has almost completed its travel, it uncovers ports in the cylinder wall, through which the burnt gases are exhausted. A current of pure air at a slight pressure enters the cylinder at this moment and sweeps them out. Thus the cylinder is freed of exhaust gas and filled with air, and is in the same condition as at the end of the suction stroke of the four-stroke cycle. During the return stroke the piston compresses the air in the cylinder, and the cycle recommences.

In other words, in the two-stroke cycle the rapid operation of the scavenging of the burnt gases, which takes place in the very short time between the end of the expansion and the commencement of the compression stroke (to which two strokes the cycle is reduced), is substituted for the two strokes of suction and exhaust of the four-stroke cycle.*

It is evident that from the thermodynamic point of view no difference exists between the two cycles described. The differences are all of a mechanical nature, and attention must be given to these if a comparison is to be made between the two methods.

At a first glance, the superiority of the two as opposed to the four-cycle might seem evident. Without altering the thermal efficiency of the engine an impulse stroke is obtained from it every revolution instead of every two revolutions of the crank shaft, and so the specific power of the cylinder should be doubled. In other terms, for equal cylinder volumes and an equal number of revolutions, a two-cycle engine would have theoretically double the power of a four-cycle engine. In addition, the mechanical efficiency is theoretically better because the two strokes of suction and exhaust, which the four-cycle engine has to carry out at the expense of the momentum of its flywheel, are eliminated. And so, for equal regularity of running, the flywheel may be less than half the weight, due to the greater regularity of the cycle itself.

In practice, the question is not so simple, nor the superiority of the two-stroke cycle so clear and evident.

First of all, two-cycle engines, unlike those of the four-stroke cycle, require a scavenging pump to supply the air for sweeping out the burnt gases. In order that the scavenging should be efficacious, this pump must be capable of giving at every revolution a quantity of air equal or greater than the cylinder volume, and needs, therefore, to be of considerable size and weight relative to the main engine proper. Moreover, it absorbs energy for its work and by the friction of its moving parts.

Secondly, the power developed by the main engine working on the two-stroke cycle is not exactly double that of a four-cycle engine of equal cylinder volume and equal piston speed, not only because of constructive reasons, but often because the imperfect scavenging leaves an appreciable percentage

* Only the modern two-stroke cycle is herein described, and no reference is made to the Lenoir cycle of gas engines, of earlier date even than the four-stroke cycle; in this case the piston drew in air and gas for a part of the outward stroke, then, all communication with the exterior being closed, the mixture was exploded, afterwards expanding to the end of the stroke. The whole of the second stroke was employed for exhaust. This cycle was naturally quite different from the four-stroke cycle, even from the point of view of thermodynamics, since there was no compression stroke.

of burnt gases in the cylinder. This is easily explained by a consideration of the limited quantity of air available for scavenging, the very short time allowed for the combined scavenging and exhaust, not to mention the impossibility of ensuring that the scavenging air shall sweep out every part of the cylinder.

A back pressure, such as may be caused by a long exhaust pipe, or one having sharp bends and not of a sufficiently large diameter, serves further to decrease the efficiency of the scavenging, causing thus a greater reduction of the power developed than the back pressure on the piston during the exhaust stroke of a four-cycle engine. For this reason, two-cycle engines require big silencers close to the engine, and exhaust pipes of large diameter.

To recapitulate, imperfect scavenging, by diminishing the percentage of oxygen in the cylinder charge, diminishes its power per unit of volume. Hence in reality, for a given size of cylinder and piston speed, the power with the two-cycle engine is not twice that of the four-cycle; this, together with the scavenging pump, alters the proportions of weight and size for the two engines, and in addition the scavenging pump reduces the advantage of increased mechanical efficiency, which would otherwise characterise the two-stroke cycle.

However, in spite of the foregoing, the superiority of the two-cycle engine is undoubted, and if there are at the present time any grounds for difference of opinion, these are merely the result of greater experience with four-cycle engines having eliminated those small difficulties which can only be revealed and remedied by practice.

Practically all makers of Diesel engines now design and build those of the two-stroke cycle, in some cases with the very best results, but always with such as to encourage continuation; it is not necessary then to claim to be a prophet to assert that the two-cycle engine is that of the future.

At present, endeavours are being especially concentrated on the design of two-cycle engines of very small and of the largest powers; and to the lesser extent on engines of medium power, for which the four-stroke cycle does not present any difficulty.

The problem is being attacked for small powers to obtain lightness and simplicity, and to give an engine sufficiently cheap for the power production of small factories, whilst for the very high powers the endeavour is to circumvent the difficulties always associated with large units, and particularly with internal combustion engines, in which large cylinder volumes give rise to anxiety regarding the strength of some parts subject to high pressure, and the cooling of those parts exposed to high temperature.

Since the thermo-dynamics of the cycle are not affected whether it has two or four strokes, it is evident that both explosion and constant-pressure engines may be constructed according to either system. There are, in fact, examples in practice of all these types, and heavy oil engines may be classified according to the number of strokes required to complete the cycle, and the manner in which combustion takes place. There are, therefore :—

Four-cycle explosion engines (semi-Diesel).
Two-cycle ,, ,,
Four-cycle constant-pressure or Diesel engines.
Two-cycle ,, ,, ,,

There are, in addition, examples of mixed combustion engines, in which the combustion takes place partly at constant volume and partly at constant pressure; naturally, these may be either of the two- or four-stroke cycle (*e.g.*, Sabathé).

Of the four categories tabulated above, in the first and second are included engines of small power (from 2 to 80 B.H.P. per cylinder) cheap in first cost and running charges; for industrial purposes those of powers less than 15 B.H.P. are the most common.

The third category is notable for the number and importance of its applications, as therein are included the numerous Diesel engines for industrial and marine purposes, both slow running and high speed, of 5 to 800 B.H.P. Those of powers less than 20 B.H.P. are not numerous, due to the relatively high first cost of the engine, which is not entirely compensated for by economical running.

The fourth division, that of the two-stroke Diesel, represents the most recent and the best application in practice of the heavy oil engine. The most powerful units (600 to 2,000 B.H.P.) are of this type, because it is naturally adapted to those powers for which the four-stroke Diesel engine, owing to its greater size, is unsuitable. This type is also used for lower powers in marine work, where the two-stroke cycle offers advantages of lightness and easier reversibility, and renders possible a greater variation of the number of revolutions.

The construction of engines of these different types may now be considered.

Four-Cycle Explosion Engines (Semi-Diesel).—The piston connected to the crank works in a cylinder closed at one end and open at the other (Fig. 29).

During the first downward stroke the piston draws air through the suction valve *a*; during the return stroke the valve *a* and every other communication with the atmosphere is closed, and the air in the cylinder is compressed; towards the end of this stroke the fuel pump P injects into the hot cast-iron bulb C (usually kept at a dark cherry-red heat) the quantity of oil necessary for the combustion stroke, so that when the piston arrives at the dead centre the fuel burns rapidly, raising the temperature and the pressure in the cylinder. During the next downward stroke of the piston, the burnt gases expand, producing useful work; during the fourth stroke the piston sweeps out the burnt gases into the atmosphere through the open valve *e*, after which the cycle recommences.

The valve *a* may be automatic, in which case the decrease of pressure due to the suction is such as to overcome the strength of the valve spring, and the valve opens due to the difference of pressure. When the suction ceases at the end of the stroke, the spring closes the valve. Frequently, however, to obtain a more certain and accurate movement, and to improve the volumetric efficiency of the engine, the suction valve is mechanically operated by a cam in the same way as the exhaust valve, which is always positively operated.

The walls and head of the cylinder are cooled by a water-jacket in order to prevent the repeated explosions from giving rise to temperatures dangerous to the strength of the walls or sufficient to burn the lubricating oil. The bulb C, on the other hand, is not cooled, as it is desired that the heat of combustion should maintain it at a temperature sufficient to vaporise and ignite fuel in contact therewith.

Naturally, before starting the engine, it is necessary to heat the bulb C, which is done by directing against it for some minutes the flame of a blow-lamp.

Generally, during the suction or compression stroke, a few drops of water or a little steam is introduced into the cylinder for the purpose of lowering the temperatures of compression and explosion. The result of this procedure is shown by practical experience to be most beneficial, though theoretically somewhat irrational.

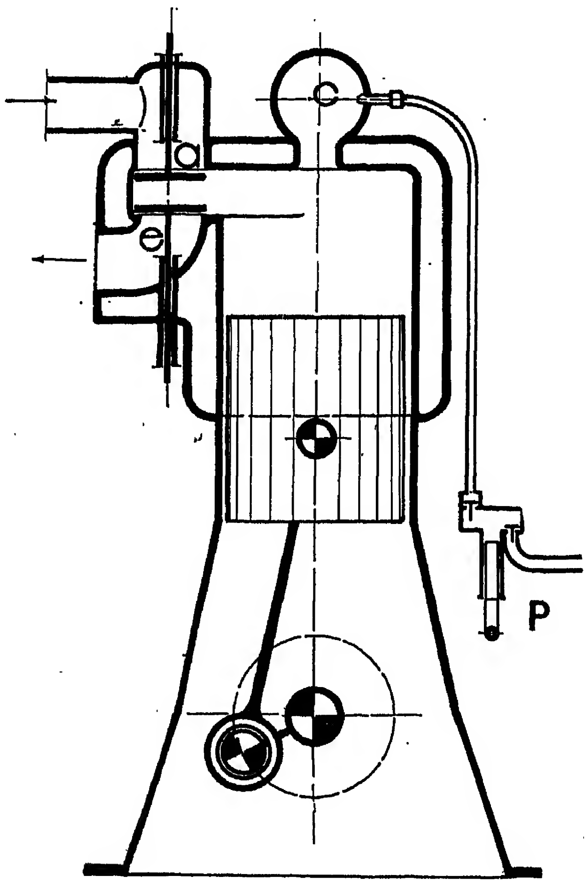


Fig. 29.—Diagram of Four-Cycle Explosion Engine (Semi-Diesel).

Four-cycle explosion (semi-Diesel) engines are made in vertical and horizontal types, and the latter resemble gas engines in appearance, and are generally of English construction.

If the term "heavy oil engines" is understood to mean those which burn a fuel with a density greater than 0·9, the examples of four-cycle

explosion engines of this class are few. Those consuming crude oils of lower density and paraffin are, however, very numerous, though not in Italy. The countries where such engines are found are determined particularly by the favourable conditions of customs duties, or the proximity to the sources of supply of the most suitable fuels. Should it be necessary to use paraffin as fuel, this type of engine is in general not the most suitable, since the paraffin may be very well vaporised in simple carburettors such as are applied to motor cars (L.W., G.F.D., Benz), eliminating the fuel pump feed and the hot bulb, the one delicate, and the other of short life, owing to the high temperature to which it is exposed.

Two-Cycle Explosion Engines (Semi-Diesel).—The most usual type has the crank-shaft totally enclosed in the crank-case, which is in communication with the exterior by an inward opening valve. The crank-case communicates in its turn with the cylinder through a passage of large size opening into the latter at such a position that the port is only completely uncovered by the piston at the lower dead centre (Figs. 30 and 31).

Another opening in the wall of the cylinder is situated diametrically opposite to, and slightly higher than, the first—*i.e.*, in such a position that the piston in its movement opens it before, and closes it after, the first. To this is connected the exhaust pipe of the engine.

Supposing the explosion to have taken place, as explained when dealing with the four-cycle engine, the piston descends under the influence of the expansion of the gases contained in the cylinder; at the same time the lower side of the piston compresses the air contained in the crank-case.

When the piston has descended far enough to uncover the exhaust ports, the gases escape through the latter to the exterior, and the pressure of those which remain in the cylinder falls to that of the atmosphere. Meanwhile, the piston descends further and uncovers the communication between the crank-case and the cylinder, through which flows the air which has been compressed in the crank-case (Fig. 30).

The piston crown is shaped in such a way as to give the current of scavenging air an upward deflection, causing it to traverse the whole cylinder before passing out through the exhaust port, driving before it and sweeping out the burnt gases which had remained in the cylinder.

On the upward stroke the piston first closes the two ports, and then compresses the remaining scavenging air, whilst its lower side draws air into the crank-case through the automatic valve to serve for the next scavenging operation (Fig. 31).

In this type of engine, as in those of the four-stroke cycle, a small quantity of water is injected into the cylinder, in order, by its evaporation, to lower the temperature at the moment of the explosion.

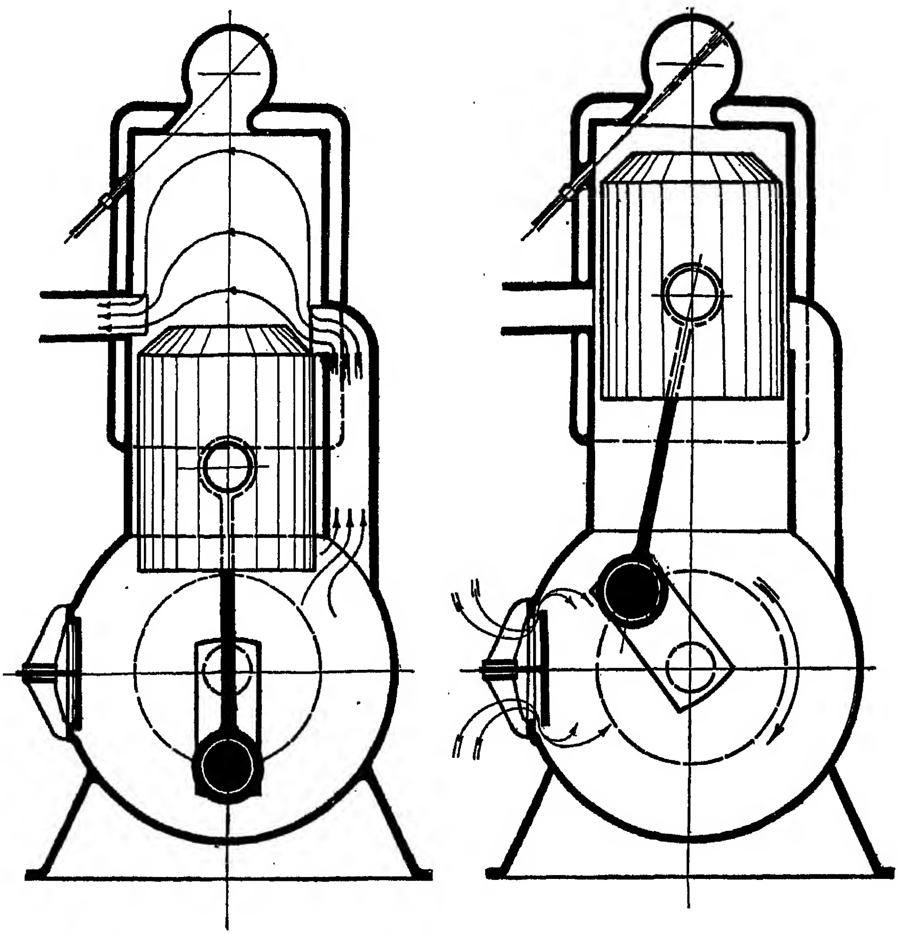
The compression pressure with these engines is 6 to 15 atmospheres (85 to 215 lbs. per square inch), and that of the scavenging air from 1 to 5 metres of water (1·5 to 7 lbs. per square inch).

The explosion pressure is from 18 to 25 kg. per square cm. (260 to 350 lbs. per square inch). The scavenging air could equally well be compressed by a special blower or by a stepped piston instead of in the crank-case, although the latter system is convenient, simple, and preferable for small powers.

Both horizontal and vertical types of these engines are constructed, though the vertical type having one, two, or more cylinders, is the more

common. Most constructors put the two types on the market, though each specialises in one or the other.

It is almost impossible to calculate and theorise regarding the construction and thermal cycle of engines of this type, as their forms and dimensions are to a great extent the result of experience, to which is due their dissimilar designs, the various pressures at which they work, and the different sections of certain parts, such as the piston crown and the hot bulb.



Figs. 30 and 31.—Diagrams of Two-Cycle Explosion Engine (Semi-Diesel).

A Four-Cycle Diesel Engine is represented diagrammatically in Fig. 32, with a single-acting piston with gudgeon pin, and all the valves *—*i.e.*,

* For simplicity in the diagrammatic sketch the valve-actuating gear is omitted. In this case, all the valves are mechanically operated.

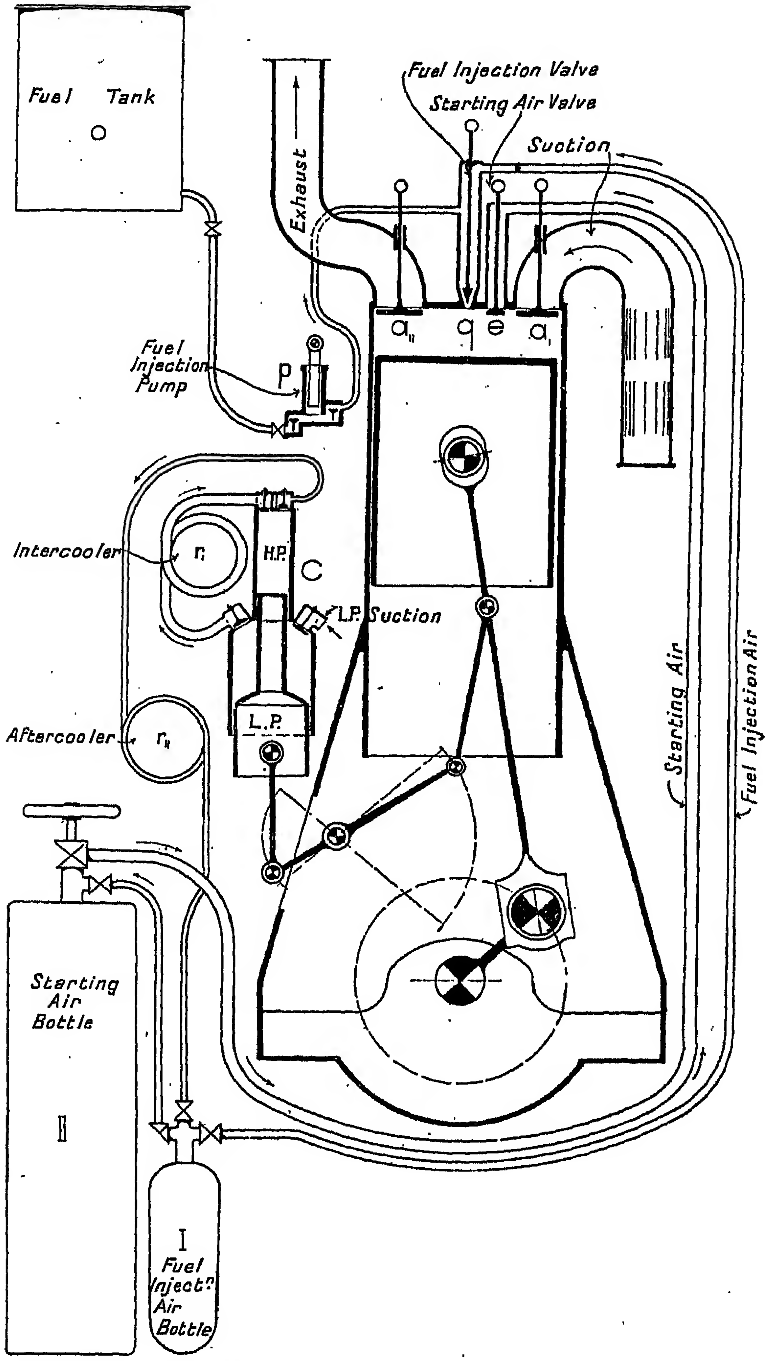


Fig. 32.—Diagram of a Four-Cycle Diesel Engine and Plant.

the inlet valve $a_{,}$, the exhaust valve $a_{,,}$, the starting air valve e, and the fuel injection valve q, in the cylinder head.

The fuel tank o supplies the fuel pump P; C is a two-stage air compressor for pumping fuel injection and starting air into the reservoirs or bottles I and II.

During the first downward stroke of the piston, air is drawn into the cylinder through the slotted pipe and the valve $a_{,}$, during the return stroke the valve $a_{,}$ and all the other communications with the exterior are closed, and the air is compressed in the cylinder to a pressure of 29 to 35 atmospheres (415 to 500 lbs. per square inch).

The compression of the air raises its temperature to such a degree that when, at the end of this stroke, the needle valve q is lifted and the fuel is injected into the cylinder, it ignites spontaneously, and the combustion proceeds approximately as long as the fuel continues to enter the cylinder—*i.e.*, a fraction of this third stroke, varying with the type of engine and with the load. The valve q is again closed, and the gases continue to expand for the remainder of the third or impulse stroke.

During the fourth stroke the piston drives out the burnt gases through the open exhaust valve $a_{,,}$.

The casing of the fuel injection valve q is always in communication with the fuel injection air bottle I containing compressed air at 45 to 70 atmospheres (640 to 1,000 lbs. per square inch), and the fuel pump P which draws the oil from the fuel tank o has to overcome this pressure during its delivery stroke. The pump P need not deliver the fuel to the injection valve at any given moment, but it must prepare in it the exact quantity for a combustion stroke. The valve q, operated by a cam, is raised at the correct moment, and allows the fuel to be injected into the cylinder by the pressure of the air in the pulveriser of the fuel injection valve. This pressure is considerably higher than that in the cylinder at the end of the compression.

Adjustments are so made that during the whole time the fuel is being injected the pressure in the cylinder is maintained practically constant, though the piston in moving increases the volume of the gases.

The compressor C supplying the fuel injection air has, as already mentioned, two or sometimes even three stages; L.P. is the low pressure and H.P. the high-pressure cylinder, whilst the coolers to lower the temperature of the air between the two stages and before it enters the reservoir are shown at $r_{,}$ and $r_{,,}$. The compressor is almost always driven by the engine itself, either by means of levers, as shown in Fig. 32, or by a crank. Its output is greater than that necessary for the fuel injection air, and the excess is stored in the starting air bottle II. There are generally three reservoirs or bottles, one similar to I and two considerably larger, as shown at II. The air from the compressor enters the reservoir I, which is connected to the fuel injection valves, and also the reservoirs II, so that when the pressure in I is sufficient for the injection valve any subsequent excess of air may be allowed to pass into the reservoirs II. When these are charged the output of the compressor is reduced to that necessary for the injection of the fuel. The piping from the reservoirs II leads to the starting air valve on the cylinder head.

A Two-Cycle Diesel Engine is represented diagrammatically in Fig. 33, in which many of the details described for a four-cycle engine, the fuel tank,

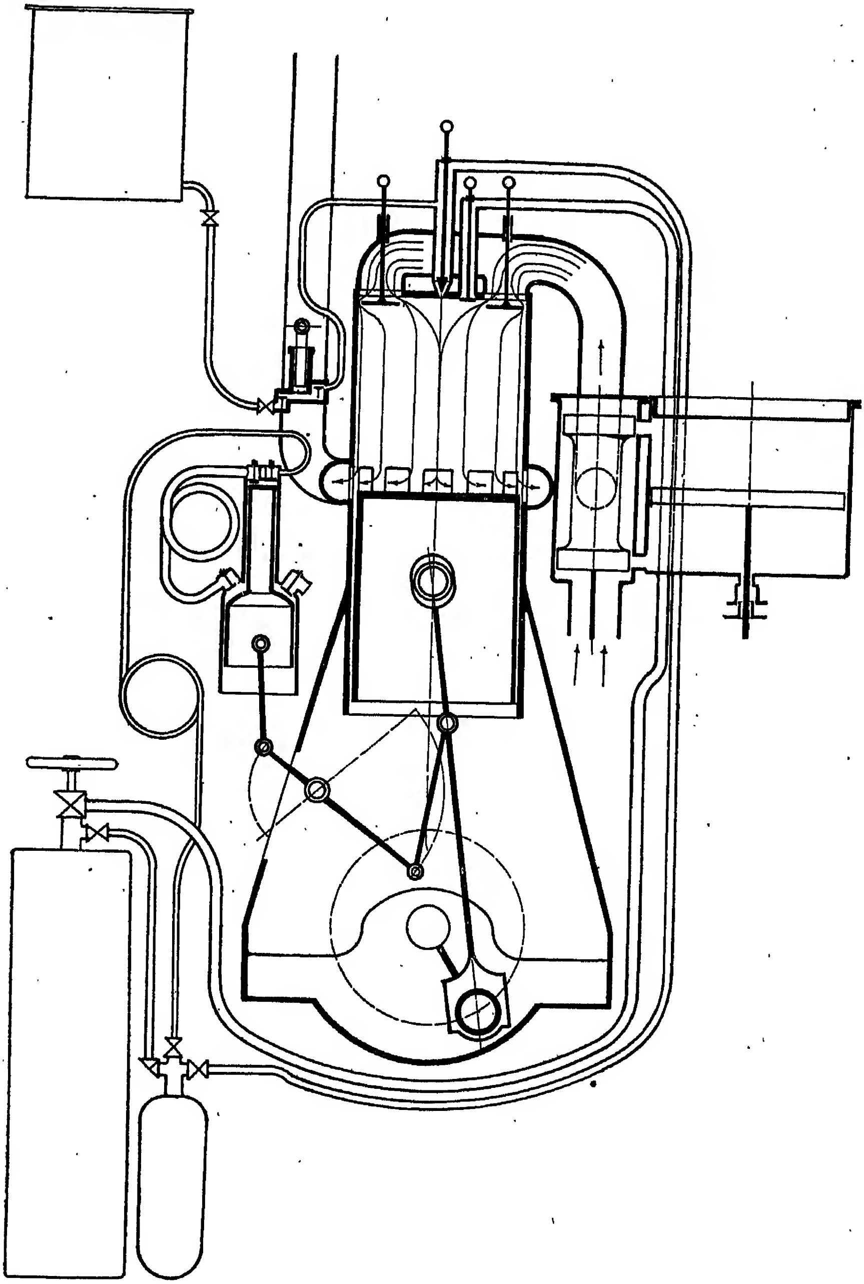


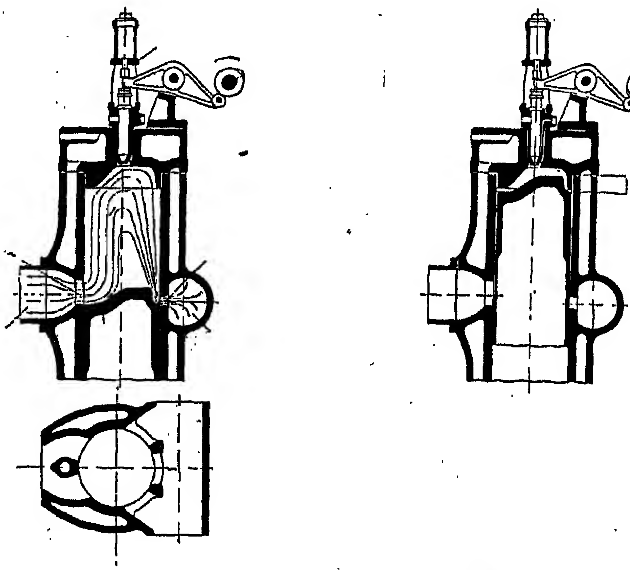
Fig. 33.—Diagram of Two-Cycle Diesel Engine and Plant.

the oil fuel pump and injection valve, the compressor with its coolers, the air reservoirs, the starting air valve, etc., will be seen.

In the lower part of the cylinder walls are seen the exhaust ports characteristic of two-cycle engines, which are uncovered by the piston at the end of its stroke.

The scavenging air, instead of entering at the bottom through other ports in the cylinder wall, in this case enters through mechanically operated valves, generally two or four in number, placed symmetrically in the cylinder head in place of the suction and exhaust valves of four-cycle engines.

When the piston has uncovered the exhaust ports, and the pressure in the cylinder has fallen to almost that of the atmosphere, the scavenging valves are opened together and allow the scavenging air to enter the cylinder and sweep out the burnt gases.



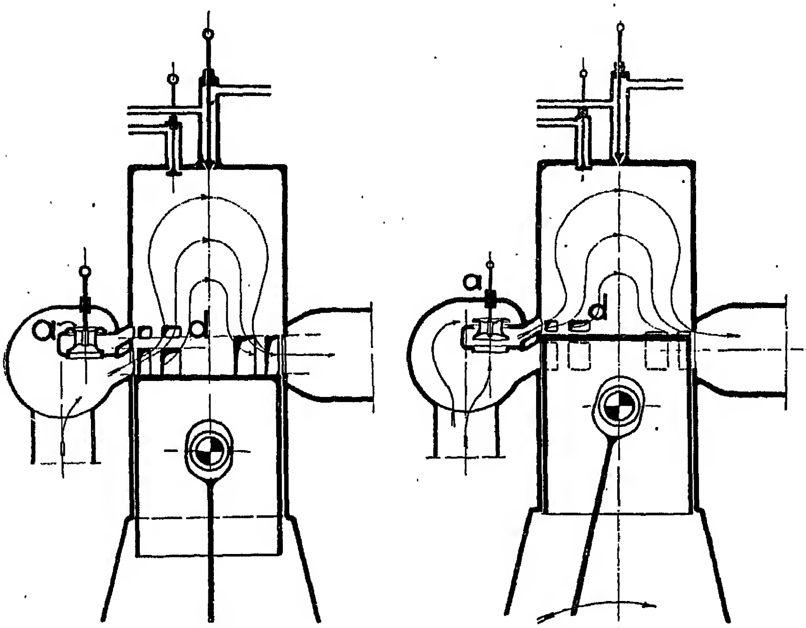
Figs. 34 to 36.—Arrangement of Port Scavenging in Polar Marine Diesel Engine.

There is nothing to prevent the scavenging being effected without valves, as described for explosion engines, and applied in some designs of Diesel engines, more especially for low powers (see Figs. 34 to 36, Polar marine engine). Scavenging with valves is more complete in its effect, and the weight of air which remains in the cylinder is greater.

When the scavenging is carried out by means of single ports, the latter are closed before those of the exhaust, and so the pressure in the cylinder is not greater than that of the atmosphere. On the other hand, with valve scavenging the valves are closed after the piston has covered the exhaust ports, and so the pressure of the air in the cylinder before the compression stroke commences is that given by the scavenging pump—*i.e.*, between 0·25 and 0·5 of an atmosphere (3 and 7 lbs. per square inch).

A system to combine the advantages of the two methods of scavenging is met with in some of the most recent two-cycle engines. Figs. 37 and 38 represent, diagrammatically, its action.

A second row of ports *d* is cast in the cylinder or liner with their upper edge higher than that of the exhaust ports. These are above the usual scavenging ports, and are opened after those for the exhaust. The ports *d* communicate with the scavenging air receiver through a separate passage, and this communication may be opened or closed by a double-seated valve *a*, operated by a cam and push rod. At the end of the expansion stroke the valve *a* remains closed and exhaust commences as usual before scavenging. On the return stroke, when the piston has covered the lower scavenging



Figs. 37 and 38.—Arrangement of Double Port Scavenging with Sulzer Marine Diesel Engine.

ports, the valve *a* opens, and air continues to flow through the ports *d*, until cut-off by the main piston, after those for the exhaust are closed. In this way the cylinder at the commencement of the compression stroke is charged with air at the pressure of the scavenging receiver, instead of at that of the atmosphere.

In two-cycle Diesel engines of high power the scavenging pump is generally independent and double-acting (Fig. 33), driven by a suitable crank from the engine crank-shaft (Tosi and Sulzer), or by balanced levers (Tosi, Carels and Germaniawerft).

The valves of the pump are often of the piston type, though sometimes

automatic valves are used instead. These scavenging pumps are generally one or two in number, even for engines of six or eight cylinders.

Instead of a separate scavenging pump, each cylinder may, to obtain a more compact engine, be provided with a piston of two diameters, the largest part of which serves as scavenging pump (M.A.N., Kind and Fiat). This system has the disadvantage that the scavenging air is hotter and, therefore, less dense, and the charge of air for combustion is thus reduced in weight.

Crank-case compression is never adopted for Diesel engines, since lack of accessibility and other disadvantages of this type of scavenging nullify

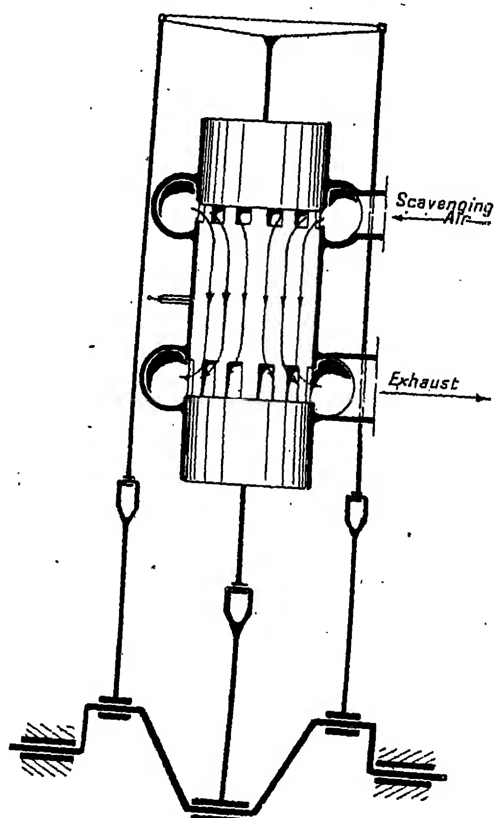


Fig. 39.—Diagram of Junkers Engine.

any gain in simplicity. It is further desirable that the scavenging pump should have a delivery greater than the engine cylinder volume.

With two-cycle Diesel engines, the compression pressure is generally from 32 to 36 atmospheres (455 to 510 lbs. per square inch), and that of the scavenging air from 0·25 to 0·35 kgs. per square cm. (3·6 to 5 lbs. per square inch), and with high-speed engines even 0·4 to 0·5 kg. per square cm. (5 to 7·1 lbs. per square inch). The fuel injection air pressure is about the same

with two-cycle as with four-cycle engines. The cylinder volume of the scavenging air pump varies from 1·3 to twice that of the engine cylinder or cylinders supplied by the pump.

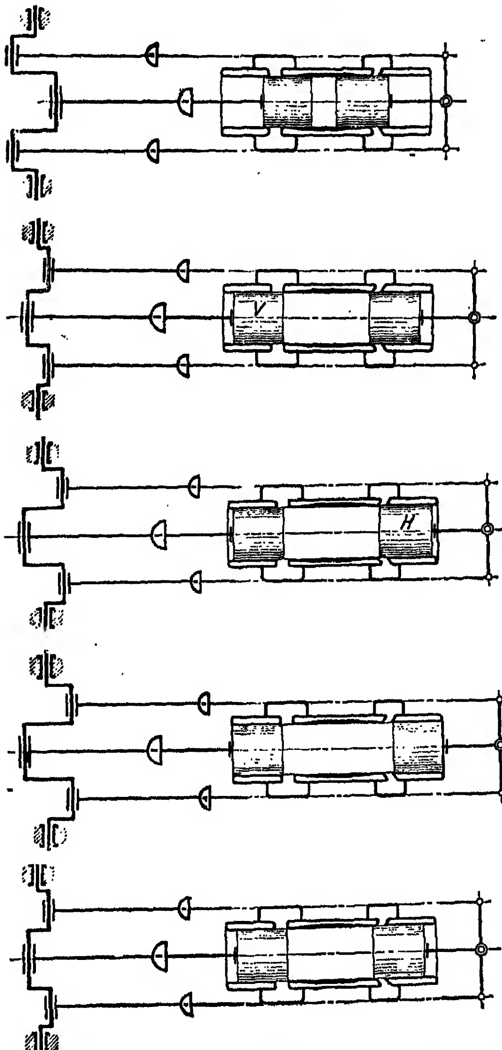


Fig. 40.

Fig. 41.

Fig. 42.

Fig. 43.

Fig. 44.

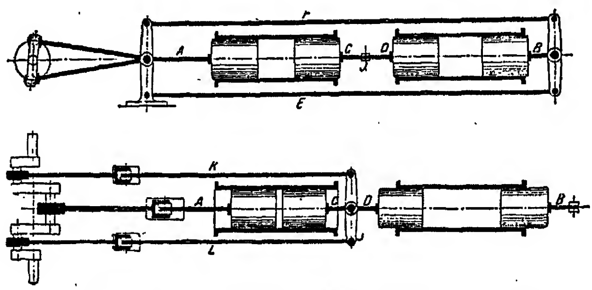
Figs. 40 to 44.—The Cycle of Operations of the Junkers Engine.

A type of two-cycle engine, which differs widely from that described hitherto is the Junkers double-piston engine evolved from the Oechelhauser gas engine. The cylinder (Fig. 39) is a simple liner without cylinder heads. In the walls are two series of ports, one for scavenging and the other for the

exhaust, covered and uncovered by the two pistons which move in opposite directions with equal strokes. When the two pistons are close to the end of the expansion stroke they uncover the ports, and the scavenging air enters at one end and drives out the burnt gases through the ports at the other end.

The engine is shown diagrammatically in Fig. 40, with the pistons at the inner dead centre. Fuel is injected into the space between the two pistons, ignites, and produces the impulse stroke, which continues until the piston V (Fig. 41) opens the exhaust ports, when the pressure in the cylinder drops to that of the atmosphere. As the pistons continue to move away from one another, the scavenging ports are uncovered by the piston H (Fig. 42) and scavenging takes place until the end of the stroke (Fig. 43), and until the pistons have on their return strokes covered first the scavenging and then the exhaust ports. During the remainder of the stroke the pistons approaching one another compress the air and the cycle recommences (Fig. 44).

Placing two such cylinders in tandem, and connecting their pistons, as shown in Figs. 45 and 46, the equivalent of a double-acting engine is obtained.*



Figs. 45 and 46.—Tandem Cylinders of Junkers Engine, equivalent to Double-Acting Engine.

The advantages of the Junkers system may be enumerated as follows :—

(1) The cylinder, being a simple liner, is well adapted to high pressures, and is not subjected to dangerous internal expansion stresses.

(2) The moving masses are in perfect balance.

(3) The reactions of the piston loads are taken by the moving parts instead of by the engine structure.

(4) The scavenging is good, and the air passes from one end of the cylinder to the other without the necessity for valves.

(5) The system is particularly suitable for high powers.

In view of the application to marine propulsion, Professor Junkers has advocated a very simple method of obtaining considerable overloads for short periods, by introducing a throttle valve in the exhaust pipe, so as to cause a back pressure in the cylinder. The scavenging pumps then automatically increase their delivery pressure, and the charge of air for combustion enclosed in the cylinder increases in weight, and more fuel oil may be burnt

* H. Junkers, *Studien und experimentelle Arbeiten zur konstrucktion meines Grossölmotors Jahrbuch der Schiffbautechnischen Gesellschaft*, 1912; *vide* also "Rivista Marittima," July and August, 1912; Schiffbau, vol. xii., 17th June, 1911.

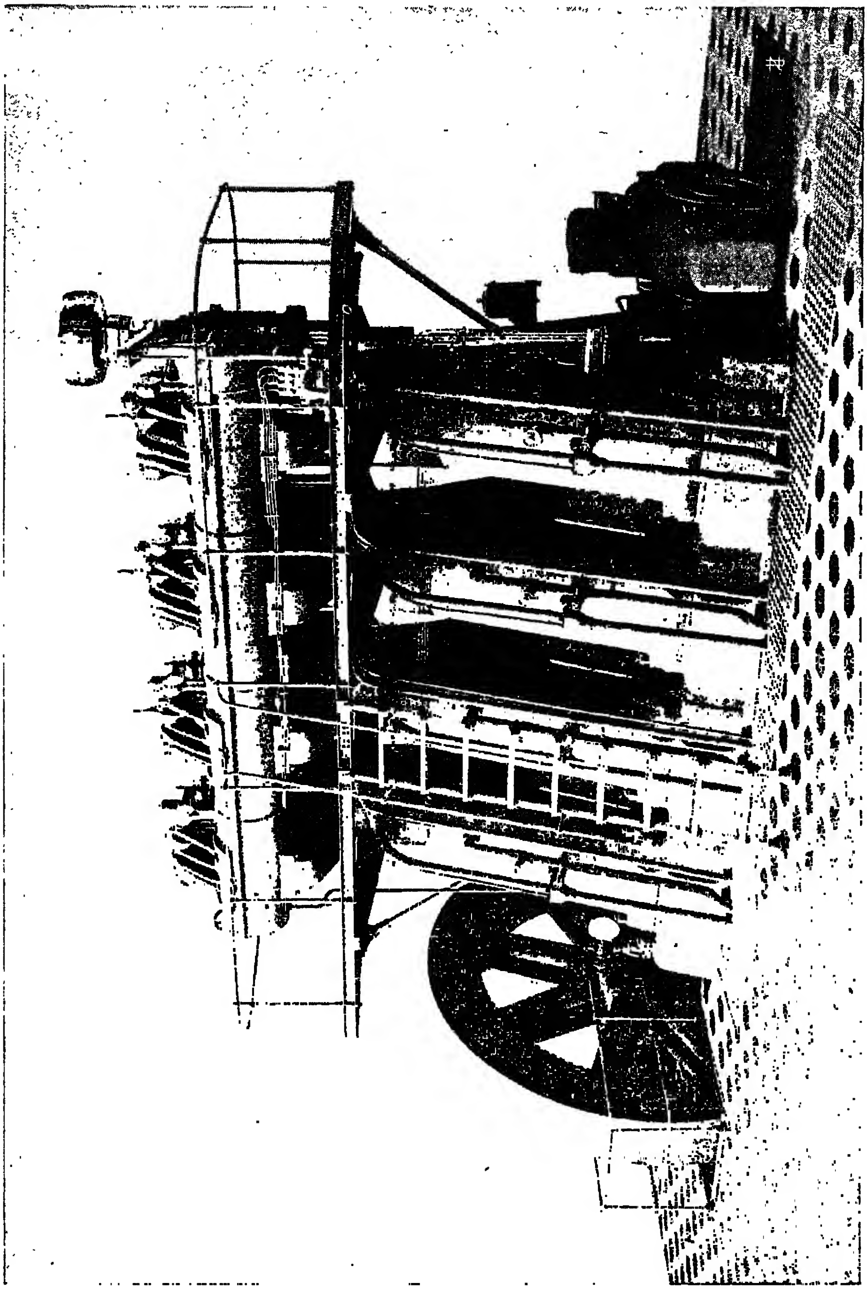


Fig. 47.—Vertical Four-Cylinder Four-Cycle Slow-Running Diesel Engine for Land Duty (Tosi Type).

to give an increase of power. This system will be dealt with later from the thermodynamic point of view (Part I., Chap. V., p. 53). It may be pointed out here that Professor Junkers easily obtains temporary increases of power up to 50 per cent. by increasing in his engine, as compared with other types, only the cylinder volume of the scavenging pump.

Stationary Engines.—The usual design of constant-pressure engines for land purposes is vertical (Fig. 47, Tosi), but during the last few years some horizontal types have been placed upon the market.

Horizontal construction offers certain advantages over the vertical type, justifying the care with which the problem has been attacked—*e.g.*, lighter and cheaper framing, greater accessibility of many of the moving parts, and, as all the actuating mechanism can be reached from the engine-room floor plates, the operations of starting and stopping, etc., are more rapid and simple than is the case with vertical engines of considerable power, where it is necessary to ascend to the engine top platform to examine the many important parts of the operating gear. The dismantling of some parts is

Fig. 48.—Twin-Cylinder Horizontal Diesel Engine (M.A.N. Type).

more simple; for example, the piston of a horizontal engine may be removed by disconnecting only the crank-pin bearing, whilst with a vertical engine it is sometimes necessary on lifting it upwards to remove first the cylinder head, together with the valve actuating levers, the suction, exhaust, fuel and compressed-air piping, etc.

The difficulties involved in the construction of horizontal Diesel engines are mainly in connection with the valve chambers and the fuel injection valve. The chief objections to this type are the greater floor space required and the difficulty of satisfactorily coupling several cylinders together. Two-cylinder engines may be built of a right-handed and a left-handed unit having the flywheel between the two, as with gas engines, or with two cylinders coupled together on a single bed (see Plates IX. and XIV. facing pp. 192 and 240, and Fig. 48, M.A.N.). With three or four cylinders it is difficult to find a good arrangement without having recourse to double-acting cylinders, as is done for gas engines. There are some

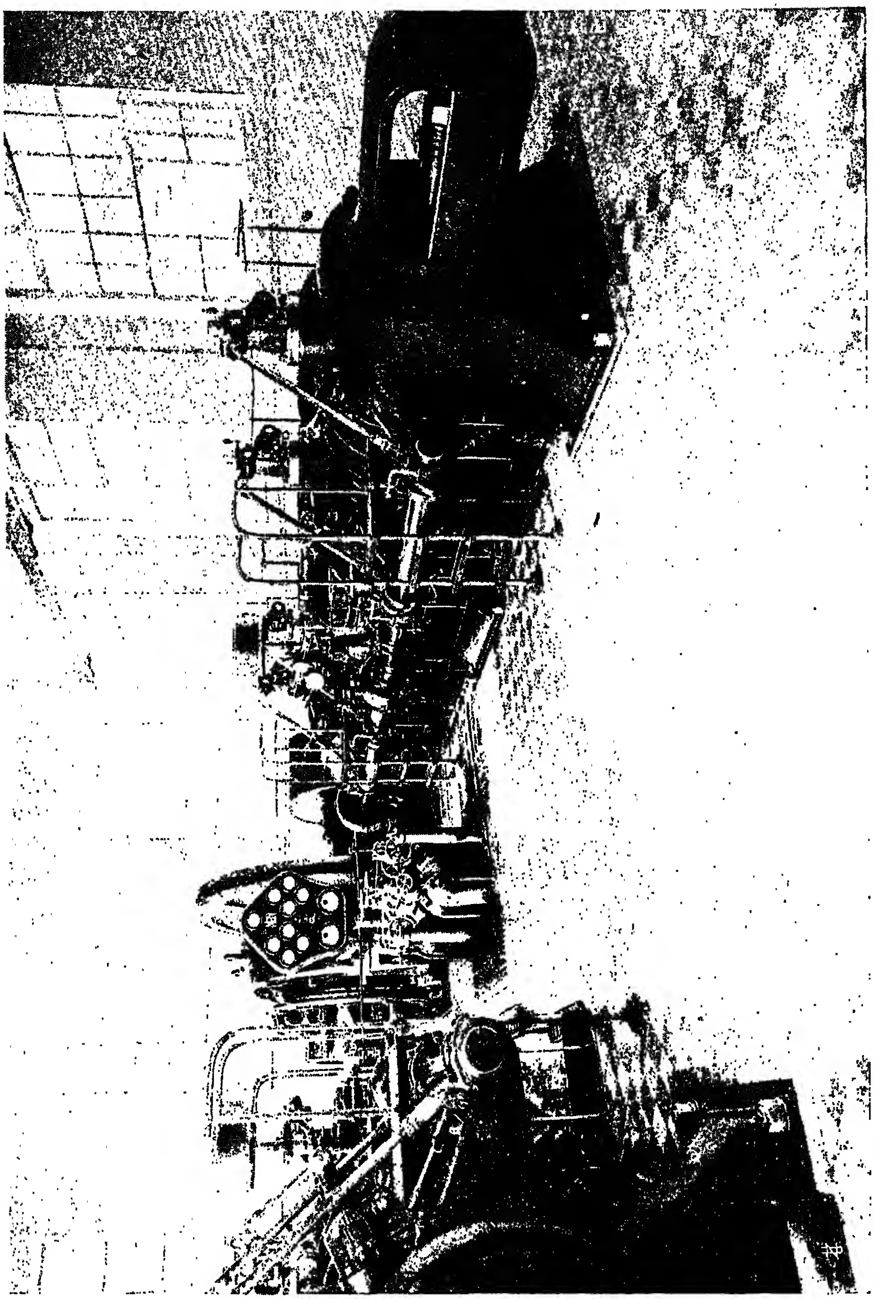


Fig. 49.—Tandem Arrangement of Horizontal Diesel Engine (M.A.N. Type).

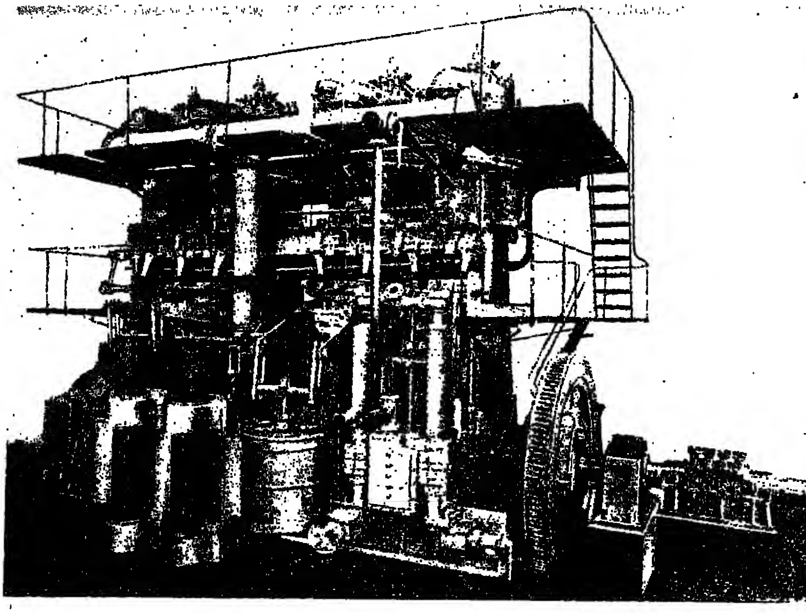


Fig. 50.—Back View of Four-Cylinder Two-Cycle Marine Diesel Engine (Carels Type).

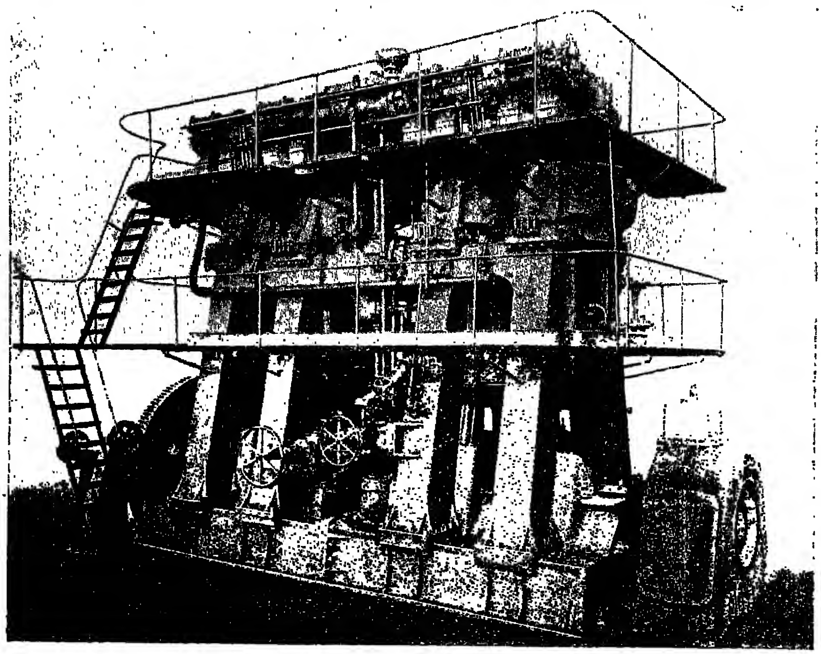


Fig. 51.—Front View of Four-Cylinder Two-Cycle Marine Diesel Engine (Carels Type).

examples of large double-acting installations in which the tandem or twin-tandem arrangement is adopted (Fig. 49, M.A.N.), as with blast-furnace gas engines.

So far, only four-cycle engines, ranging to about 500 B.H.P. per cylinder, have been referred to, but some of the largest firms are designing or have on trial two-cycle engines of even 1,000 B.H.P. per cylinder, and capable of giving, when arranged as a twin-tandem group, an output of 4,000 B.H.P.

Marine Engines.—Marine Diesel engines are always vertical (Figs. 50 and 51, Carels), with two, three, four, or even eight cylinders in line. Only with the Junkers system is the horizontal type proposed.

The grouping of a number of cylinders, besides permitting of high powers without the difficulties due to very large cylinder volumes, reduces the height of the engine (an important condition, especially with warships), diminishes vibration, and permits of a greater variation of the number of revolutions.

A good marine engine, for manœuvring in and out of harbour, etc., ought to be able to run with perfect regularity and without danger of stopping even at very slow speeds.

The two-cycle double-acting marine engine of 6,000 B.H.P. of the firm M.A.N. Nürnberg, is the most powerful so far constructed.*

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* In 1912 a fire caused by a partial explosion during a shop trial seriously damaged this engine.

CHAPTER IV.

THERMO-DYNAMIC CYCLES.

In the preceding chapter the general methods of working of heavy oil engines have been enumerated and described, and the manner in which the heat of the fuel is transformed into useful work pointed out.

The cycles will now be more closely analysed. All the theory and general calculations for such engines are based on a study of these cycles, and for that reason it is necessary, briefly, to re-examine those questions having direct practical application.

The theoretical work diagram (having volumes as abscissæ and pressures as ordinates) of a four-cycle explosion engine is shown in Fig. 52. The curve

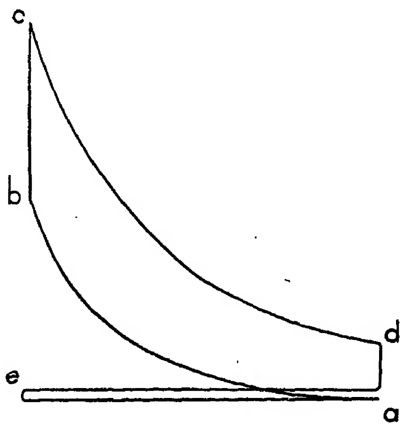

Fig. 52.—Theoretical Diagram of a Four-Cycle Explosion Engine.

ab is that of compression, *bc* represents the combustion at constant volume, *cd* the expansion, *de* the exhaust, and *ea* the suction.

If the engine works on the two-stroke cycle the suction stroke is eliminated, and the exhaust and scavenging are represented by the line *dfa* (Fig. 53).

Fig. 54 represents the diagram of a four-cycle engine with combustion at constant pressure; the compression *ab* continues to the maximum pressure of the cycle, *bc* is the combustion at constant pressure, *cd*, *de*, and *ea* are respectively the expansion, exhaust, and suction.

Again, if the two-stroke cycle is considered, the line *dfa* (Fig. 55) is substituted for *d*, *e*, *a*.

These two cycles are by far the most frequent in practice, but in addition there is the cycle of mixed combustion as applied with Sabathé engines, in

which the only variation is that the combustion takes place in two phases; one *bc* at constant volume, followed by another $cc_{,}$ at constant pressure (Fig. 56).

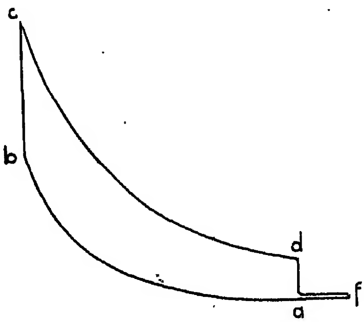

Fig. 53.—Theoretical Diagram of Two-Cycle Explosion Engine.

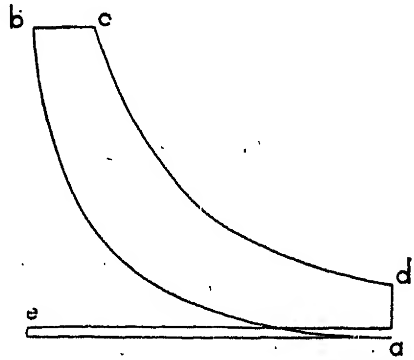

Fig. 54.—Theoretical Diagram of Four-Cycle Constant Pressure Engine.

The Suction of Four-Cycle Engines.—At the commencement of the suction stroke, the volume v_1 comprised between the top of the piston and that of the cylinder (the combustion space) is full of the burnt products of combustion.

The piston commences to move outwards and draws in air through the open suction valve. The pressure in the cylinder, which was that of the atmosphere, or a little more, is lowered to a value p_a, a little less than the atmosphere. This depression is due to the loss of pressure when the air is drawn through the suction pipes and valve with a velocity consequent upon the volume generated by the piston movement in any given time.

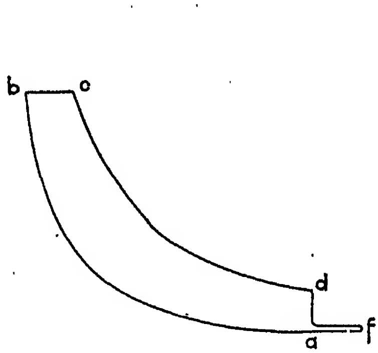

Fig. 55.—Theoretical Diagram of Two-Cycle Constant Pressure Engine.

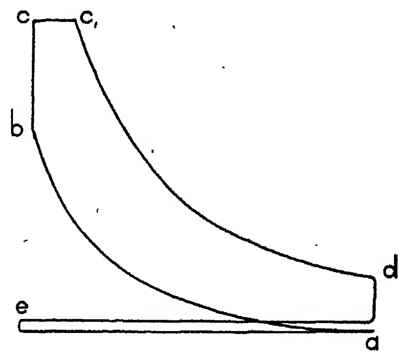

Fig. 56.—Theoretical Diagram of Mixed Combustion Engine (Sabathé Type).

If the area and the lift of the valves are suitably proportioned to the dimensions and speed of the engine, the suction stroke is regular, and the pressure varies but little for the different piston positions.

The weight of suction air for the complete stroke is given by :—

$$G_a = \frac{p_a \cdot v}{T_a \cdot R_a} = \eta_v \frac{10{,}000 \cdot v}{T_a \cdot R_a},$$

in which $p_a = \eta_v \times 10{,}000$, and is the pressure of suction, T_a the absolute temperature of the air, R_a the constant for air, and v the cylinder volume.*

In practice generally η_v, which, as will be seen, represents the volumetric efficiency of suction, is about 0·9 for slow-running engines and about 0·85 for those of high speed, but is considerably lower with engines of the highest speed with automatic inlet valves—opening under the influence of pressure difference instead of being positively operated by levers. T_a varies generally from 290° to 300° C. absolute.

To obtain the weight of gas in the cylinder at the end of the suction stroke, the weight of the burnt gases which are in the combustion chamber at the commencement of the suction stroke must be added. These have a pressure p_e = about 1·10 atmospheres (15·6 lbs. per square inch), and a temperature T_e = about 700° to 800° C. absolute, and occupy the volume v_1 of the combustion space. The total weight of the gas in the cylinder is then

$$G = \frac{v_1\, p_e}{T_e \cdot R_e} + \eta_v \frac{v\, 10{,}000}{R_a \cdot T_a},$$

in which R_e can be obtained when the composition of the exhaust gas is known.

Exhaust of Four-Cycle Engines.—In the phase of the exhaust two distinct periods occur; firstly, the gas escapes rapidly at the end of the expansion stroke, by virtue of its pressure being higher than that of the atmosphere; and, secondly, it is pumped out of the cylinder by the incoming piston.

In the first period, the pressure drops rapidly from p_d to p_e in a very short time, so that from the theoretical point of view this drop may be considered as taking place at constant volume (although this does not correspond exactly to practical conditions).

The velocity of the gas is from 700 to 800 metres (2,300 to 2,600 feet) per second, and its temperature, T_d, 900° to 1,300° C. absolute.

In the second period the mean value of the pressure p_e may be taken at about 1·1 atmospheres (15·7 lbs. per square inch); 1·08 (15·35 lbs. per square inch) for slow-running engines, and 1·15 atmospheres (16·35 lbs. per square inch) for those of high speed. The rules governing this second period are, however, somewhat irregular and fluctuating, depending on the efflux velocity of the gas in the first period.

The temperature of the exhaust gas measured in the vicinity of the exhaust valve is from 600° to 700° C. absolute, and varies with the piston speed and with the conditions, good or bad, of combustion.

---

* In this and all the following formulæ the suffix letters applied to the symbols indicate the point of the theoretical diagrams shown in Figs. 52 to 56, to which they refer. For example, P_e and T_e indicate the pressures and the absolute temperatures respectively of the points e at the end of exhaust of the diagrams, P_d and T_d the pressures and the temperatures at the commencement of exhaust, etc.

Exhaust and Scavenging of Two-Cycle Engines.—It has already been stated that with two-cycle engines the exhaust and suction strokes are eliminated, and in their place scavenging by means of a current of air is substituted. This air enters the cylinder through valves or ports, and part of it escapes through the exhaust ports cast in the cylinder liner or barrel, pushing before it the burnt products of combustion, and leaving the cylinder charged with air as pure as is possible.

The exhaust ports are uncovered slightly before the valves or ports for the scavenging air are opened, and in this period of the first phase of exhaust the pressure drops to p_a—slightly greater than one atmosphere (14·22 lbs. per square inch). After this, the scavenging commences, and the period of scavenging corresponds to the exhaust stroke of four-cycle engines.

Since the exhaust ports must be uncovered by the piston before those for scavenging, in the cases where the scavenging air enters by means of ports, it follows that the latter must be closed before those for exhaust (see Figs. 30 to 34). If, however, the air enters by means of valves (see Fig. 33) it is arranged that these close some time after the piston has covered the exhaust ports. This second system, as explained in the preceding chapter, has the advantage that at the end of the scavenging period the cylinder is full of air at a pressure greater than that of the atmosphere, and thus the weight of combustion air per unit volume is augmented.

Whilst the weight of air drawn in by a four-cycle engine is

$$\frac{p_a \cdot v}{T_a \cdot R_a},$$

the charge of air in a two-cycle engine of equal cylinder volume v is

$$\frac{p_l\,(v + v_1)}{T'_a \cdot R'_a},$$

where p_l is the pressure at the end of scavenging and v_1 the volume of the combustion chamber. Supposing

$$T_a = T'_a \text{ and } R_a = R'_a,$$

and making the ratio between the two charges,

$$\frac{p_l \cdot (v + v_1)}{p_a \cdot v} > 1,$$

it is seen, since p_l is always greater than atmospheric pressure whilst p_a is lower, that $v + v_1 > v$.

For this reason, the quantity of air contained in unit volume of a two-cycle cylinder, and consequently the power developed for equal piston speeds, should be more than double (the number of combustion strokes per second is double) that of a four-cycle engine. In practice, this is not the case; the current of scavenging air cannot form a perfect piston; it has different velocities and directions from point to point, and assumes a vortical or whirling motion at the periphery of the column, neglecting the most remote corners of the cylinder, for which reason the percentage of oxygen in the charge is so much less than with a four-cycle engine as to falsify the results of the preceding considerations.

Combustion.—In explosion engines combustion should theoretically take place at constant volume, and the line *bc*, which represents it in the diagrams (Figs. 52 and 53), should be parallel to the ordinate.

In practice, the time taken for combustion has an actual value, however small; and the explosion appears on the diagram as a line very slightly curved towards the expansion line.*

It may be added that, generally, during the first part of the expansion period, the burning of a part of the fuel continues, giving rise in this way to the phenomenon known as gradual combustion, on which subject several important treatises have been written, notably those of Witz, Clark, Otto, Tresca, etc.

The rapidity, duration, and completeness of this gradual combustion are influenced by the ignition system, the volatility and purity of the fuel, and the temperature of the cylinder and the air—*i.e.*, by the compression ratio.

Supposing instantaneous ignition (Figs. 52 and 53), then

$$p_c = \frac{p_b \cdot T_c}{T_v} = \frac{T_c \cdot G_a \cdot R_a}{v_1},$$

$$T_c = \frac{p_c \cdot v_1}{G_a \cdot R_a},$$

in which T_c often reaches 1,500° to 2,000° C. absolute.

With combustion at constant pressure the theoretical line *bc* (Figs. 54 and 55) is parallel to the abscissæ. Practically, however, this condition cannot be obtained, as it is impossible to inject the fuel into the cylinder in such a way that combustion takes place exactly at constant pressure. In practice it almost always happens that the pressure rises a little at the beginning of injection and falls more or less during the remainder of the phase.

Theoretically,

$$p_c = p_b, \quad T_c = T_b \frac{v_1}{v_2} = T_b \cdot \rho,$$

where v_1 is the volume ot the combustion space, and v_2 that of the combustion space + the volume generated by the piston in moving from *b* to *c* of the combustion period of the stroke. The ratio $\frac{v_1}{v_2} = \rho$, and is called the full pressure ratio.

With mixed combustion the part of the curve *bc* at constant volume is identical with that of the explosion cycle, and the period at constant pressure *c c*, commences at the pressure and temperature reached at the point *c* of the curve of constant volume.

Compression and Expansion.—The following remarks under the heading of these two thermo-dynamic transformations hold good equally for two- or four-cycle engines of the explosion or constant-pressure types.

The form of their curves has the formula $Pv^k = \text{const}$

Although, in all calculations, k is regarded as constant throughout these changes, actually this does not represent the real thermo-dynamic conditions.

The value of k depends on the exchange of heat which takes place from

* If combustion occurs too early—*i.e.*, before the piston has reached the inner dead centre—the combustion line may incline towards the zero ordinate.

the working fluid in the cylinder to the retaining walls. Since the temperature of the fluid varies throughout the stroke continuously, and differently from the variations of the wall temperatures, the value of k varies with different engines, and is influenced even in the same engine by the temperature of the cooling water, the load, the degree of tightness of the piston and of the valves.

The exponent in the equation to the curve has a mean practical value of $k = 1{\cdot}30$ to $1{\cdot}35$ for compression and $k = 1{\cdot}35$ to $1{\cdot}5$ for expansion.*

For theoretical considerations, the curves are taken as adiabatics—*i.e.*, it is assumed that the walls of the cylinders neither absorb nor give up any heat.

The following equations serve for the calculation of the compression volume to obtain a given maximum pressure :—

$$p_b = p_a\left(\frac{v + v_1}{v_1}\right)^k = p_a \cdot \varepsilon^k$$

$$T_b = T_a\left(\frac{p_b}{p_a}\right)^{\frac{k-1}{k}} = T_a \cdot \varepsilon^{k-1},$$

in which the ratio $\frac{v + v_1}{v_1} = \varepsilon$, and is called the compression ratio.

The following formulæ serve for the expansion curve :—

With explosion engines—

$$p_d = p_c\left(\frac{v_1}{v + v_1}\right)^k,$$

$$T_d = T_c\left(\frac{v_1}{v + v_1}\right)^{k-1}.$$

With constant pressure engines—

$$p_d = p_c\left(\frac{v_2}{v + v_1}\right)^k,$$

$$T_d = T_c\left(\frac{v_2}{v + v_1}\right)^{k-1}.$$

To draw the curves, the Brauer method (Fig. 57) may be used; oa is drawn making any angle α with oX, and to draw ob, the angle β is determined so that $(1 + \tan\beta) = (1 + \tan\alpha)^k$.

From any point of known co-ordinates $p_o v_o$, two lines are drawn perpendicular respectively to oX and oY. From the points where these perpendiculars cut the axes oY and oX, lines are drawn to ob and oa to make

---

* Instead of using a mean value for the exponent of v in the equation, the curve can with much closer approximation be divided into parts, each of which follows a different law as regards the exponent. Prof. Belluzzo advises a division into three parts for the expansion curve, with $k = 1$ to $1{\cdot}2$ from the point of maximum pressure to that of a pressure $0{\cdot}8\ p_c$; $k = 1{\cdot}4$ to $1{\cdot}45$ between $0{\cdot}8\ p_c$ and $0{\cdot}5\ p_c$; and $k = 1{\cdot}45$ to $1{\cdot}5$ between $0{\cdot}5\ p_c$ and the end of the stroke.

an angle of 45° with oY and oX. From the points where these lines at 45° intersect ob and oa, new perpendiculars to oY and oX are drawn, and their point of intersection gives another point in the desired curve.

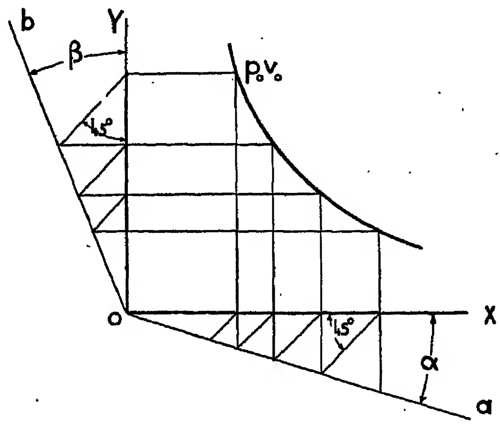

Fig. 57.

The following table gives values for β for most practical cases :—

k =	1·10	1·15	1·20	1·25	1·30	1·35	1·41	1·50
α	11° 20′	11° 20′	11° 20′	14° 05′	14° 05′	14° 05′	18° 25′	18° 25′
β	12° 35′	13° 10′	13° 50′	17° 55′	18° 40′	19° 25′	26° 30′	28°
Tan α	0·20	0·20	0·20	0·25	0·25	0·25	0·33	0·33
Tan β	0·222	0·234	0·245	0·322	0·337	0·352	0·497	0·540

CHAPTER V.

EFFICIENCIES.

THE efficiencies to be considered in an internal combustion engine are as follow :—

The Mechanical Efficiency—the percentage of unit power developed in the cylinder, which can be taken from the engine at the flywheel. The difference between this percentage and unity is employed to overcome the friction of the moving parts. Mechanical efficiency is, therefore, the ratio between the brake horse-power and that indicated by the diagram—*i.e.*,

$$\eta_m = \frac{N_e}{N_i}.$$

Indicated and Brake Thermal Efficiencies (η_{ti}, η_{te})—the percentages of the heat units of the fuel that the engine is capable of transforming into indicated or effective work ; in other words, the ratio between the equivalent of one H.P. in heat units and the number of heat units which the engine requires to develop one I.H.P. or one B.H.P.,

$$\eta_{ti} = \frac{75 \times 3{,}600}{425} \times \frac{N_i}{P . V} = \text{about } 635 \frac{N_i}{P . V},$$

$$\eta_{te} = \frac{75 \times 3{,}600}{425} \times \frac{N_e}{P . V} = \text{about } 635 \frac{N_e}{P . V},$$

where P is the weight of fuel in kilogrammes consumed in an hour with a calorific value of V calories per kg.

The Volumetric Efficiency in a four-cycle engine—the ratio between the weight of the air contained in the cylinder at the commencement of the compression stroke and that required to fill the same volume with air at atmospheric pressure.

The Scavenging Efficiency in a two-cycle engine—the ratio between the weight of air in the cylinder at the commencement of the compression stroke and that of the mixture of air and burnt gases.

The mechanical efficiency depends on the type of engine, and on the quality, quantity, and system of lubrication. The speed of revolution has a certain influence, as also the temperature of the cooling water, for should this be too low, the friction between the piston and the cylinder may be augmented,* due to the alteration of clearance between the piston and cylinder walls.

* Cavalli, *Teoria del motore a scoppio* (Napoli, 1911), p. 72.

New engines have always a slightly lower mechanical efficiency than those that have run for some time, since actual running gives better surfaces than can be obtained by machining, however well this latter may be carried out.

The mechanical efficiency varies also with the load, and

$$\eta_m = \frac{N_i - N_v}{N_i},$$

in which N_v is the horse-power absorbed by the friction of the engine, which remains practically constant whatever may be the power developed for a given speed of revolution.

It has been seen, when dealing with cycles, that the expansion is never continued to atmospheric pressure, and the exhaust always commences when the gas has a pressure of 2 or more atmospheres (28·5 lbs. per square inch or over). If the engine friction be considered as a pressure P_f referred to the area of the piston, the work developed in that part of the expansion curve between $p = P_f$ and the end of expansion is seen to be less than that absorbed by friction.

To give a value to mechanical efficiency, an indicator diagram has to be worked out (assuming that the card is sufficiently accurate), and the result so obtained compared with the reading of the brake or other power measuring apparatus taken at the same time.

Mechanical efficiency varies as a rule with heavy oil engines from 0·70 to 0·85 at full power. With a good four-cycle engine of medium or high power, the efficiency often reaches 0·80; in those of the two-stroke cycle, on account of the friction of the scavenging pump and the larger size of the compressor, the efficiency seldom exceeds 0·70.

Whilst the mechanical efficiency depends almost entirely on the construction of the engine, the thermal efficiency is closely related to the thermodynamics of the cycle.

Theoretically the explosion cycle is considered as consisting of two adiabatics *ab* and *cd* (Fig. 52) and of two changes at constant volume *bc* and *da*, whereas the constant pressure cycle is taken to be two adiabatics *ab* and *cd* (Fig. 54) with a change at constant volume *da* and a constant pressure curve *cd*.

Both diagrams (Figs. 52 and 54) are the outcome of practice and not of theory, and do not tend to approach to the Carnot cycle of maximum efficiency composed of two adiabatics and two isothermals.

In the "*Ideal Wärmemotor*" originally studied by Diesel, a cycle approaching that of Carnot was suggested, but apart from the difficulty of attaining isothermal changes with a fluid of low specific heat, such as a gas, the original cycle demanded pressures and temperatures inadmissible in practice.

In reality the first engine suggested by Diesel, and described in the now famous treatise "Theory and Project of a Rational Heat Engine destined to supplant steam engines and heat engines already in use," should have burnt coal dust. With this engine, at the end of adiabatic compression a pressure of 250 atmospheres (3,560 lbs. per square inch) should have been reached, and at the end of isothermal combustion 90 atmospheres (1,270 lbs. per square inch). The cylinder instead of having a water jacket should have been insulated to prevent any dispersion of heat. The thermal efficiency

ought to have reached 73 per cent. The first engine (Fig. 58) was constructed

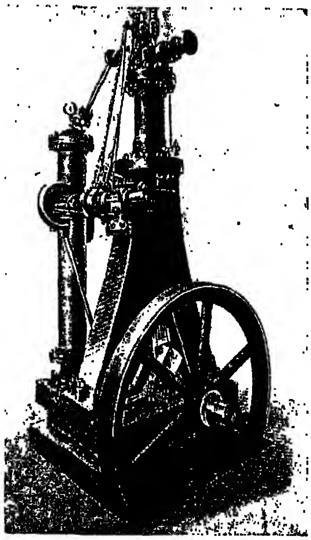

Fig. 58.—First Experimental Diesel Engine to be designed for Coal Dust as Fuel.

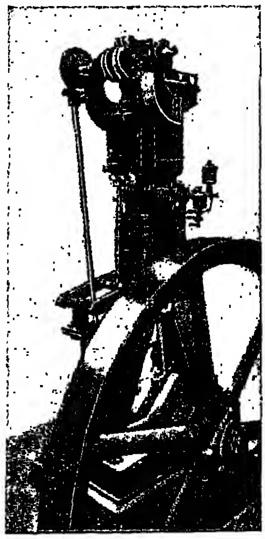

Fig. 59.—Diesel Type Engine evolved in 1895 from Experiments.

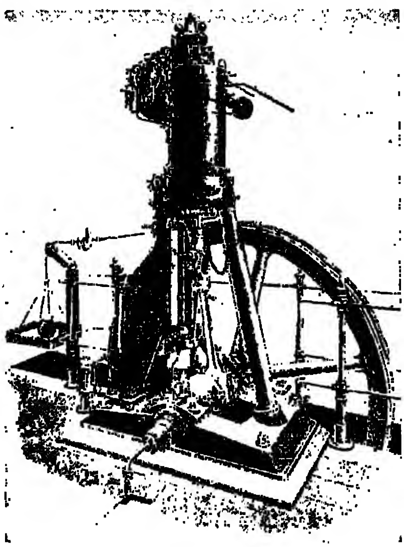

Fig. 60.—Engine first publicly tried in 1897.

at the works of the Maschinenfabrik, Augsburg, in financial collaboration with the firm of Fried. Krupp of Essen, and was used largely for experiments

carried out by Diesel and Engineer L. Vogel (Prof. von Lossow, *Zeit. des Ver. deutsch. Ing.*, No. 27, 1903).

After this, various procedures for combination and many types of fuel were tested, and after mechanical and thermo-dynamic modifications, a type very similar to present-day practice, and represented in Fig. 59, was arrived at in 1895, and later that shown in Fig. 60, with which the first public experiments were carried out in 1897 by Prof. Schröter. This engine is now preserved in the German Museum at Munich (Room 13).

The first Diesel engines were exhibited to the public at the Munich Exhibition of 1898. There were three, one from the Maschinenfabrik, Augsburg, one from Krupp, and the third from the Gasmotorenfabrik Deutz. From this point the commercial application of the Diesel engine commenced.

The general expression for thermal efficiency in internal combustion engines is

$$\eta_t - 1 - \frac{\varphi_2}{\varphi_1} = \frac{\varphi}{\varphi_1},$$

in which φ_1 is the heat contained in the fuel, φ_2 the heat lost, and φ that actually used. This expression is changed with explosion engines, if the compression and expansion are assumed as adiabatics, into

$$\eta_t = 1 - \frac{T_a}{T_b},$$

in which T_a and T_b are the absolute initial and final temperatures of compression; or, in other words, *the thermal efficiency is augmented with an increase of the final temperature of compression.*

But since,

$$\frac{T_a}{T_b} = \left(\frac{v_1}{v + v_1}\right)^{k-1} = \frac{1}{\varepsilon^{k-1}},$$

and

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}},$$

we see that *the theoretical thermal efficiency of an explosion engine increases as the compression ratio, or with the final compression pressure.*

The efficiency increases also, though to a lesser degree, with an increase in the value of *k*—*i.e.*, of the ratio of the specific heats of the gas at constant pressure and at constant volume—and, since the value of *k* increases with weak mixtures, the *weak mixture high compression rule* is justified. This rule forms the key to most of the improvements effected with explosion engines from their inception to the present time.

To the advantage as regards thermal efficiency conferred by high compressions, should be added that of easier and quicker ignition of the combustible charge and the smaller quantity of burnt gases which the mixture contains, due to the reduced volume of the combustion chamber.

Practical considerations prevent the compression pressure from being raised indefinitely, and the advantages of going beyond 16 to 20 atmospheres (230 to 290 lbs. per square inch),* are negligible.

* Güldner.

For Diesel engines the general expression—

$$\eta_t = 1 - \frac{\varphi_2}{\varphi_1} = \frac{\varphi}{\varphi_1}$$

is changed to

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \times \frac{\rho^k - 1}{k(\rho - 1)},$$

in which

$$\rho = \frac{v_1 + v_e}{v_1},$$

where v_1 is the volume of the combustion chamber and v_e that generated by the piston in the part *bc* of the diagram, during which combustion at constant pressure takes place.

The value ρ, as explained in the preceding chapter, is called the *ratio of full pressure.*

In this case the efficiency is raised, as with explosion engines, with an increase of the compression, but is influenced inversely by the value of the *ratio of full pressure.**

In this way it is seen, and is confirmed by experience, that the thermal efficiency of Diesel engines is greater at reduced loads, where the value of ρ is smaller than at the maximum power.

This condition gives one of the most appreciable advantages of the Diesel engine—*i.e.*, the small increase in fuel consumption per unit power at reduced loads. The increased thermal efficiency compensates, in part, for the loss of mechanical efficiency (Fig. 62).

* From the formula which gives η_t it is seen that the thermal efficiency remains constant when adopting the Junkers system for obtaining a large overload, as applied to engines of the type having opposed pistons. In this system the engine exhaust is throttled and the scavenging pumps have to overcome a greater pressure, and so introduce the charge of air into the cylinder at higher pressure. The ratio between the exhaust pressure and the final compression pressure remains the same, and also the value ρ may be constant (Fig. 61).

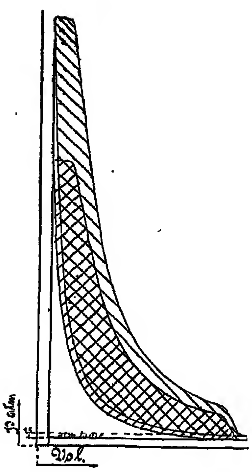

Fig. 61.—Indicator Diagrams of Junkers Engine, with Exhaust Free and Exhaust Throttled.

The explosion and constant pressure cycles have been deeply studied and compared with the aid of scientific examples, and have led on occasions to very heated arguments. Diesel, Zeuner, Schröter, Schöttler, Witz, Güldner, Banki, etc., have discussed and written at length on this question. At the present time, the general opinion is that the two cycles are more or less equivalent, and that the thermal efficiency of the Diesel engine depends

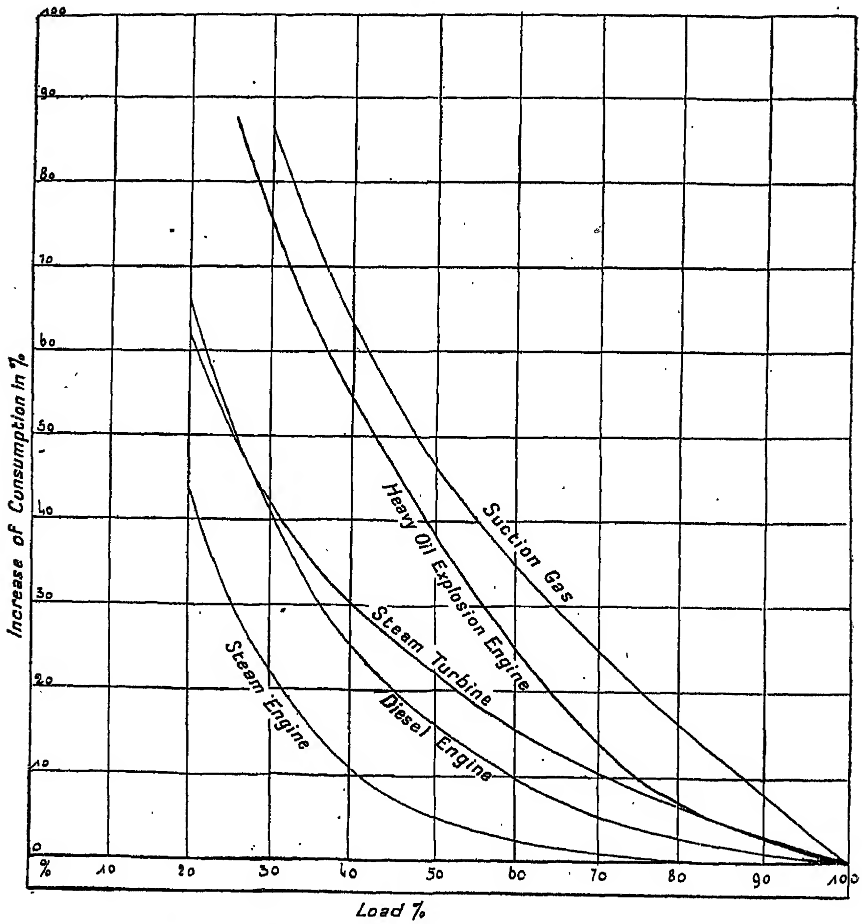

Fig. 62.—Diagram illustrating Comparative Efficiencies of Diesel and other Engines.

upon the favourable and practical conditions of working which cannot be attained in other cycles (including the cycle with isothermal combustion).*

The general problem—given a quantity of heat, h, to find the thermodynamic conditions of heat introduction in the cycle that the efficiency may be a maximum—is obviously indeterminate. Comparison between the cycles

* Schöttler, *Die Gasmachine*, 4th ed., p. 269.

cannot be made without fixing the pressure, but may be made by taking equal compression pressures or equal maximum pressures.

The first method, probably the more rational from the theoretical standpoint, leads to a conclusion in favour of the explosion cycle.*

The second, which has a greater practical value by reason of the limiting conditions of strength, and working clearances of the engine parts tending to keep the maximum pressure within certain bounds, results in a conclusion in favour of the cycle at constant pressure.

A clear and simple comparison can be made with the aid of the theoretical entropy diagram as given by Boulvin (Fig. 63).

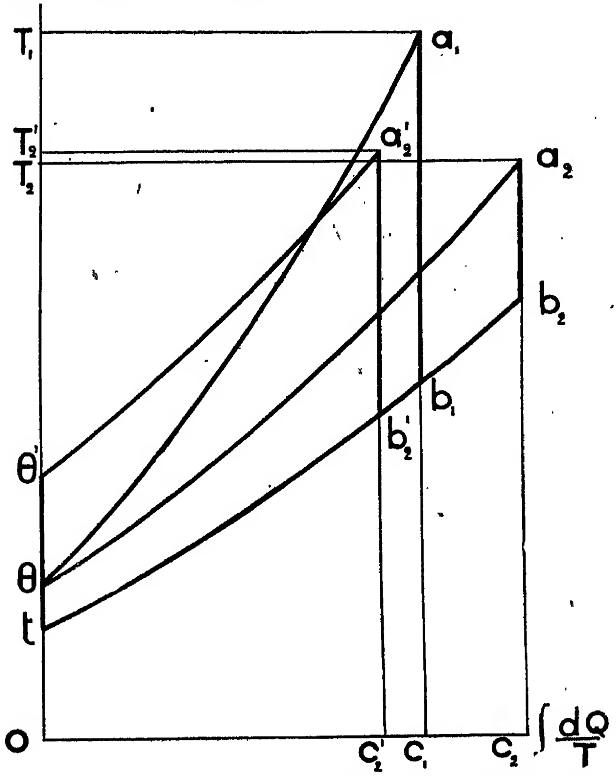

Fig. 63.—Theoretical Entropy Diagram of Explosion and Constant Pressure Engines.

In the entropy diagrams, as seen, the ordinates are the absolute temperatures T and the abscissæ the entropy $\int\frac{d\varphi}{T}$. The area $\int\frac{d\varphi}{T} \times T = \varphi$ represents the quantities of heat.

The diagram $t\theta a_1 b_1$ (Fig. 63) is that of an explosion engine; t is the temperature at commencement and θ that at the end of compression; the compression curve, if the change is considered as adiabatic, is the curve $t\theta$, parallel

* Clearly explained in the article by Prof. Meyer, *Zeit. des Ver. deutsch. Ing.*, 1897, p. 1108.

to the ordinate. The combustion at constant volume is θa_1, and $a_1 b_1$ is the adiabatic of expansion, and $b_1 t$ that of exhaust. The heat used is $t\theta a_1 b_1$ and that expended $o\theta a_1 c_1$. The ratio between these two areas gives the efficiency.

To construct the diagram for the cycle at constant pressure, commencing with the same conditions at the end of compression and expending the same quantity of heat, the curve $t\theta$ remains that of compression, θa_2 is the combustion at constant pressure, $a_2 b_2$ is the expansion, and $b_2 t$ the exhaust.

If it is desired that the heat expended be the same for the two cycles the area $o\theta a_1 c_1$ should equal $o\theta a_2 c_2$, but, since $otb_2 c_2$, the heat lost with the constant pressure cycle is greater than $otb_1 c_1$ (that lost with the explosion cycle), it is evident that *for equal degrees of compression the explosion cycle is superior to that at constant pressure.*

By taking advantage, as is done in practice, of the possibility offered by the Diesel engine of carrying compressions to much higher values than are used with explosion engines, the foregoing conclusion may be reversed. Considering the cycle at constant pressure $t\theta' a_2' b_2'$, commencing at the end of compression with a temperature θ' greater than θ, since the heat expended is still $o\theta' a_2' c_2' = o\theta a_2 c_2 = o\theta a_1 c_1$, whilst the heat lost is $otb_2' c_2'$, which is less than that lost with other cycles, the contention is confirmed that *the thermal efficiency of the constant pressure cycle is superior in practice to that of the explosion cycle, since, in the former, compression can be carried to higher values.*

To examine with accuracy the thermo-dynamic values of the various cycles, recourse must be had to the entropy temperature diagrams; moreover, it is a simple matter to convert the diagram of work as given by the indicator card into an entropy temperature diagram. The entropy of the gas is

$$\mathrm{E} = \int \frac{d\varphi}{\mathrm{T}} = c_v \, . \log_e pv^k + \text{const.}$$

Since the product pv is proportional to the absolute temperature, the values of this product may be taken to any given scale as ordinates of the diagram.

If, then, the variations of entropy, instead of its absolute values, be taken, the constant of the equation may be eliminated and the entropy becomes: *

$$\mathrm{E} = \log_e p + k \log_e v.$$

When the values obtained from this formula have been used to draw the diagram, the scale may be found by comparing the planimetered area of the diagram with the heat units used—always a known quantity—and by taking the ratio between any known temperature and that which corresponds to it in the diagram. The lowest temperature of the cycle can also be assumed as unity and all the others referred thereto.†

To change a curve $\mathrm{P} = f(v)$ into the form $\mathrm{T} = f(\mathrm{E})$ by graphical methods, the procedure originated by Ancona‡ and quoted by Schöttler may be used.

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* If c_2 remains a constant.

† Schöttler, *Die Gasmachine*, 4th edition, p. 273.

‡ Ancona, *Graphische Theorie der Otto-Gasmaschine.* Verh. f. Gewerbfleiss, 1895, p. 334.

Let AB (Fig. 64) be the curve $P = f(v)$, then by selecting R as unit of temperature, $pv = T$ is obtained, as already stated. By drawing from the points P of the curve horizontal lines which cut the vertical drawn from the abscissa 1, and drawing also from O radii which cut in T the verticals let fall from the points P, a curve of points such as T will be obtained and will have absolute temperatures as ordinates.

If the logarithmic curve $\log v = f(v)$ be now drawn and the ordinates be multiplied graphically by k, the curve $k \log v = f(v)$ will result. By regarding the natural logarithms of the pressures as negative ordinates, the vertical distance between the curve of these and $k \log v = f(v)$ will be

$$k \log v + \log p = E,$$

that is, the abscissæ of the required curve $T = f(E)$.*

In the construction of these complete diagrams, the following points should be borne in mind:—The cycle should be closed, the points of an adiabatic should all lie in a vertical line, and the horizontal distance between two constant volume curves or between two constant pressure curves is such that, having drawn one of these curves and a point on the other, it is not necessary to carry out all the construction to complete the diagram.

Stodola has suggested a method for drawing any curve given a constant pressure and a constant volume line.†

The heat units φ_2 which are lost in the cycle of internal combustion engines are principally given to the cooling water, or are to be found in the exhaust gases. A mean figure for each of these losses, with heavy oil engines, is approximately 28 to 30 per cent. of the total heat of combustion of the fuel. Generally, with engines of a high piston speed the greater loss is in the exhaust gases, since these have not had sufficient time to give up much of their heat to the cylinder jackets, and so they escape from the cylinder at a relatively high temperature. On the other hand, with slow-running engines, the greater loss is due to the heat given to the cooling water. Experimentally, it is easy to evaluate both of these losses, but it is useful sometimes to estimate the approximate values for each for a given engine, more especially if it is intended to use a portion of this lost heat for useful work (see p. 6).

For the heat lost to the cooling water, Cavalli ‡ has proposed the following formula based upon the investigations of Fourier :—

$$\frac{V_r}{V} = \beta \cdot \frac{T_c - \theta}{1{,}000} \times \frac{\dfrac{1}{R} + \dfrac{\varepsilon + 1}{\varepsilon - 1}}{\eta_v \sqrt{Dn}},$$

in which V_r is the heat dispersed in the water, V the calorific value of the fuel, β the coefficient which has a value of about 1·30, T_c the temperature of combustion, and θ that of the water. R is the ratio between the diameter D of the cylinder and the stroke, n the number of revolutions per minute, ε and η_v the ratio of compression and the volumetric efficiency respectively.

* Schöttler, *opus cit.*, p. 274.

† Stodola, " Die Kreisprocesse der Gasmaschinen," *Zeitsch. des Ver. deut. Ing.*, 1898, p. 1045.

‡ Cavalli, *Teoria del motore a scoppio* (Napoli, 1911), p. 75.

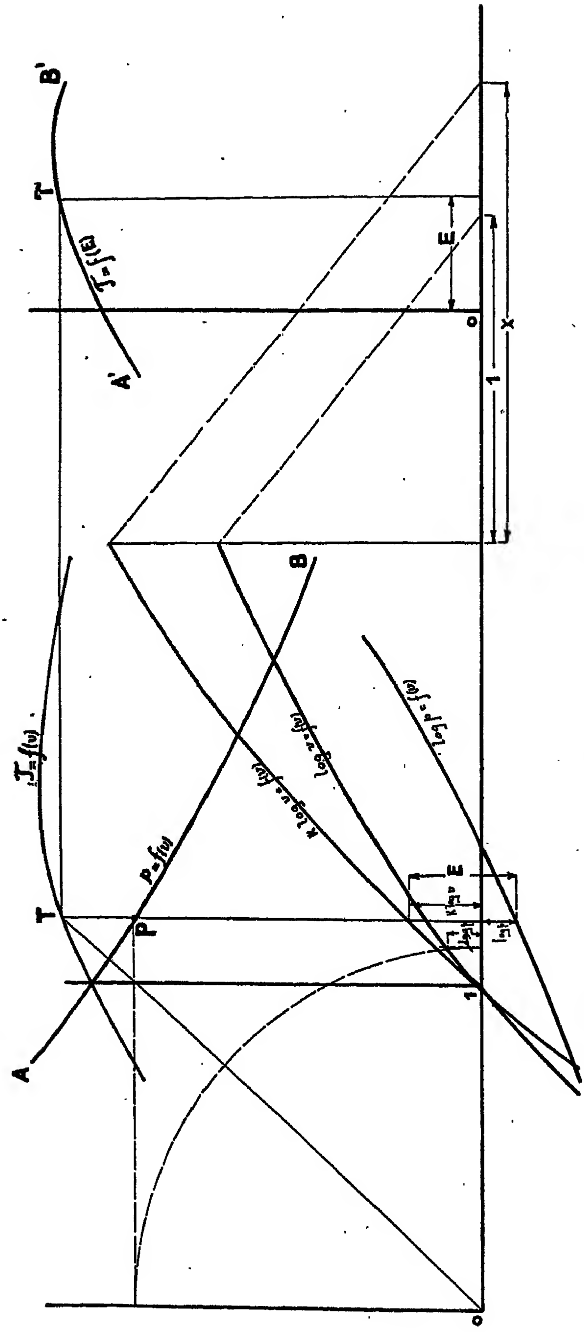

Fig. 64.—Graphic Conversion of Curve T = f (v) into Curve T = f (E) (Ancona).

The calculation for the heat carried away by the exhaust gas is very simple.

The heat lost is $V_s = C_p . G . T$, in which C_p is the specific heat of the exhaust gas (which is obtained from their chemical composition and is about 0·24), T is their temperature, which varies from 550° to 750° C. absolute according to the type of engine, and G is the weight of the exhaust gases, which is merely the summation of the weight of suction air, fuel, and fuel injection air.

In practice the effective thermal efficiency of Diesel engines is generally above 30 per cent. at full power.

In fact, assuming 185 to 230 grammes (0·41 to 0·51 lb.) per B.H.P.-hour to be the limiting consumptions for large and small engines respectively, and that the fuel oil has a calorific value of about 10,000 calories (18,000 B.Th.U. per lb.), the total heat units φ_1 given to the engine are always between the following values :—

$$0{\cdot}185 \times 10{,}000 = 1{,}850 \text{ calories},$$

and

$$0{\cdot}230 \times 10{,}000 = 2{,}300 \text{ calories}.$$

The heat units required per B.H.P.-hour are—

$$\varphi = \frac{75 \times 3{,}600}{425} = 635 \text{ calories (2,540 B.Th.U.)},$$

whence

$$\eta_{te} = \frac{\varphi}{\varphi_1} = \frac{635}{1{,}850 \text{ or } 2{,}300} = \text{from } 0{\cdot}35 \text{ to } 0{\cdot}28.$$

In Fig. 65 the consumptions per B.H.P. per hour of various types of heat prime movers are given, and it is clearly to be seen that the Diesel engine, whether of the two- or four-cycle type, consumes a smaller number of heat units per B.H.P.-hour than any other prime mover, and for that reason has the highest thermal efficiency.

The Volumetric Efficiency depends entirely upon the type of engine, the design of valve gear, piston speed, and the openings of the valves or ports, but is independent of the thermo-dynamic transformations throughout the cycle.

With four-cycle engines the volumetric efficiency η_v is measured approximately by the drop of pressure in the cylinder during the suction stroke :—

$$\eta_v = \frac{p_a}{10{,}000}.$$

Naturally the importance of this drop of pressure is not to be attributed to the slight negative work produced by such depression (similar to that resulting from the exhaust stroke), but rather to the smaller quantity of combustion-supporting-air which is in the cylinder at the commencement of the compression stroke, and which is responsible for a lowering of the power developed per unit of cylinder volume and per unit of time.

In dealing with this question, it may be mentioned that the height of the engine above the sea level has an influence on the volumetric efficiency, and so upon the power which an engine of given dimensions is capable of developing (see p. 7).

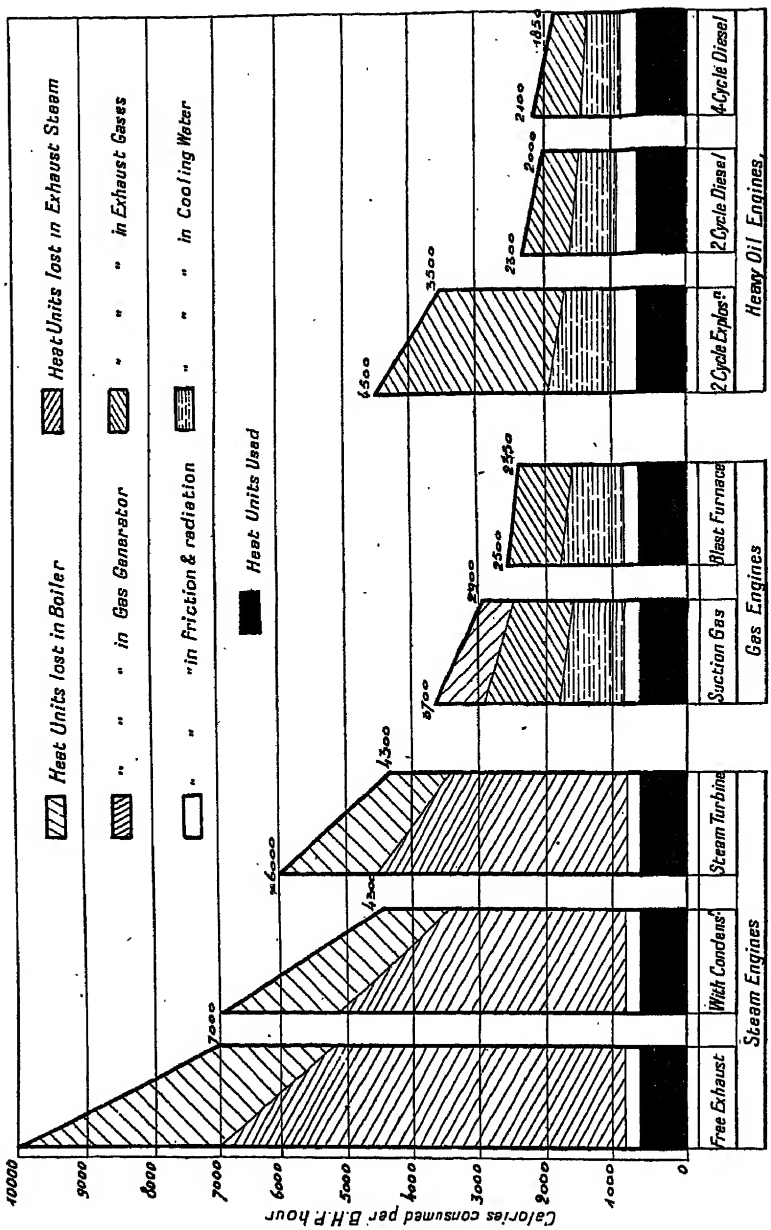

Fig. 65.—Comparative Diagram of Heat Consumptions per B.H.P. per hour of various types of Heat Prime Movers.

Scavenging Efficiency.—The phase of scavenging with two-cycle engines is difficult to subject to a process of sufficiently simple, general and approximate calculation, which will serve any useful and practical purpose.

Scientific examinations of this rapid operation, of vital importance to the overall efficiency of the engine, are rarely met with in those well-known treatises, which for the most part were published before the two-stroke cycle engine had the decided industrial application which it has to-day.

Professor Meyer, in a treatise dealing with the methods to be applied to scientific research on scavenging,* proposes to assume a temperature of 15° C., and a pressure of 1 kg. per square cm. (14·2 lbs. per square inch) as the normal conditions, and calls the *scavenging efficiency* the ratio between the pure air that remains in the cylinder at the commencement of the compression stroke (this air is some fraction of the total delivery of the scavenging pump) and the total contents of the cylinder at the same moment, which consists of a mixture of scavenging air and residual air and burnt gases—the two latter remaining from the previous combustion stroke. *The efficiency of the charge* is defined as the ratio between the amount of pure air in the cylinder at the commencement of compression and the total cylinder volume, whilst *the useful scavenging effect* is the ratio between this same amount of pure air and the total output of the scavenging pump.

The values which may be assigned to these efficiencies for various types of engines are not known at present, and the above treatise of Professor Meyer is only to propose a method for research, which, on account of its scientific accuracy and simplicity, will no doubt receive in the near future ample treatment at the hands of experimenters.

* P. Meyer, "Gründlagen für die Untersuchung von Zweitaktmaschinen," *Zeitsch. des Ver. deut. Ing.*, 5th October, 1912, No. 40.

CHAPTER VI.

THE CALCULATION OF CYLINDER DIMENSIONS.

THE problem connected with the determination of the cylinder diameter and piston stroke, for a given engine power, is always somewhat uncertain on account of the coefficients, depending upon current practice and experimental results which enter into the calculation, and owing to the necessity of making *a priori* certain assumptions of primary importance.

In the formula

$$75\, N_e = 10{,}000 \,.\, \eta_m \,.\, \frac{1}{v} \,.\, p_m \,.\, \frac{\pi \,.\, D^2}{4} \,.\, \frac{2nS}{60}, \quad . \quad . \quad . \quad (1)$$

η_m is the mechanical efficiency of the engine, p_m the mean pressure of the indicator diagram, D and S are the diameter and stroke of the piston respectively, n the number of revolutions per minute, and v the number of strokes required to complete the cycle.

All the dimensions are in metres and the pressures in kgs. per sq. cm.

Taking for four-cycle engines $v = 4$ and grouping the constants, the formula becomes

$$N_e = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, n \,.\, S, \quad . \quad . \quad . \quad (2)$$

or, making $v = 2$ and grouping the constants in the same way for two-cycle engines, the equation is

$$N_e = 1{\cdot}743 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, n \,.\, S. \quad . \quad . \quad . \quad (3)$$

Values for η_m and p_m are assigned as a result of experience with the type of engine under consideration, and, naturally, the greater that experience the nearer the approximation.

In order to solve the equation, as is usually done, in terms of the cylinder diameter D, values must be given to the number of revolutions per min. n and the piston stroke S.

Instead of fixing separately the number of revolutions per min. and the stroke of the engine, a mean piston speed $\frac{2 \,.\, n \,.\, S}{60}$ for a given speed of revolution may be taken, or a given stroke bore ratio and an assumed value for one of the other quantities, such as the revolutions or piston speed, may be proposed for solving the equation.

Generally several trials are made, varying the values of n and S and the results of the substitution compared with existing data of similar engines.

The number of revolutions is often one of the specified factors, and when not given, is always approximately indicated by the type of engine (fast or slow-running, marine or stationary, for continuous or intermittent-running, etc.), as well as from its power, and whether two- or four-stroke cycle.

From the formula (1), p. 66, it is seen that the power of the engine is proportional, within practical limits, to the speed of revolution. It is not true, however, that an engine of 100 B.H.P. at 200 revolutions per minute will develop 200 B.H.P. at 400 revolutions per minute; the increasing speed of revolution gives a more imperfect indicator diagram, the higher speeds of suction or scavenging air diminish the volumetric efficiency, the back pressures are greater, the friction increases, and, due to the increased speed, the time allotted to any given phase is shorter. This latter consideration makes for inefficiency, since certain factors, such as the speed of the suction air, depending only upon differences of pressure, do not increase in proportion to the increased speed of revolution. Before commencing the design of an engine, the speed of revolution must, of course, be known, and the fact that the efficiency diminishes with high-speed engines as compared with those of lower speed must be taken into consideration.

Greater importance than is necessary is sometimes given to the values of the stroke-bore ratio $\frac{S}{D}$. From the thermal standpoint, this ratio only affects the ratio of the surface of the cooled walls of the cylinder to the volume of the same.

Research up to date gives information regarding the minimum surface for dispersing the necessary percentage of the heat of combustion, and whilst this data should not be neglected, its importance must not be exaggerated to the exclusion of more important considerations.*

From the point of view of construction, the ratio $\frac{S}{D}$ influences the height of the combustion chamber.

Short strokes necessitate, for equal compressions, smaller clearances between the head of the cylinder and the crown of the piston at the top

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* The research carried out by Cavalli (*Motori a Scoppio*, Naples, 1911, p. 58) gives the formula

$$\frac{S}{D} = 2\left(\frac{\epsilon - 1}{\epsilon + 1}\right),$$

which for engines using high compression, approaches too nearly to the value $\frac{S}{D} = 2$, and would give an excessive piston speed to high-powered or high-speed engines.

The value of this formula is negatived by the fact that Cavalli in his calculations only allowed for the time during which the various internal surfaces of the cylinder were in contact with the gases, and not for the various mean temperatures of the gas corresponding to the various points of the stroke.

Moreover, identical conditions are assumed for the cylinder and piston crowns. This is far from being the case, especially with air-cooled pistons.

Prof. Belluzzo ("Notes on Internal Combustion Engines," *Industria*, 1911-12), in a similar research carried out on a more practical basis, assigns coefficients to the various heat dispersing surfaces, which, although not depending solely upon general calculation, succeed in approaching more closely to the actual values than do those of Cavalli.

Belluzzo arrives at the formula

$$S^3 = \frac{2N_e}{Kn},$$

in which K is the coefficient in the formula, $N_e = K \cdot D^2 \cdot S \cdot n$, dealt with in this chapter (p. 70).

Results from this formula approximate sufficiently nearly to the values of S used in practice.

dead centre; with high compressions this may cause difficulty in the design in order to obtain the required clearance with a suitable arrangement and lift of the valves; moreover, small differences in this height, due to inexact workmanship, would have an appreciable influence on the final compression pressure.

The piston speed is of primary practical importance, and although satisfactory theoretical explanations to justify this fact are lacking, it may be stated that there is a speed which gives, in practice, the best results for every type of engine. In general, the highest efficiency is obtained with the lowest piston speeds; this statement is not absolutely exact, and the same speed cannot be assigned to engines of greatly varying power. The mean speed of the piston always increases with the power, although the greater the power of the engine the lower the speed of revolution, in order to avoid obviously disproportionate values for the ratio $\frac{S}{D}$.

The inertia forces are, of course, dependent upon the piston speed.

For stationary Diesel engines the following values may be substituted in the general formula, as representing usual practice :—

η_m = 0·75 to 0·8 for slow-running four-cycle engines.
= about 0·7 for high-speed four-cycle engines.
= about 0·7 for slow-running two-cycle engines.

p_m = 6·5 to 7 kgs. cm.^{2} (92 to 100 lbs. per square inch) at normal loads, according to the type of engine and its speed of revolution; this value permits of a temporary overload of 20 per cent. The value of p_m can be better estimated by means of the rules laid down in the preceding chapter for the theoretical diagram by fixing the duration of combustion, and introducing a coefficient to allow for the imperfections of the indicator diagram obtained in practice.

$v = \frac{2 \, . \, n \, . \, S}{60}$ = 3 metres (9·85 feet) per second for land engines of small and medium power, up to 4 metres (13·2 feet) per second for the largest powers in use; for high-speed engines 3·5 to 4·5 metres (11·5 to 14·75 feet) per second or even more is used.

n = 250 to 150 revolutions per minute for small to high powers for land engines.
= 375 to 250 revolutions per minute for small to high powers with high-speed engines.

Examples.—The examples which follow are from recent land engines of well-known firms, and serve to help in the choice of values for the coefficients.

(1) *Four-cycle Diesel Engine of* 25 *B.H.P.* $n = 220$.

Putting v = about 3 metres (9·85 feet) per second,

then $S = \frac{30v}{n} = \frac{30 \times 3}{220}$ and S = about 0·409.

And so S = say 0·40 m.

Then $N_e = 25 = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, 220 \times 0{\cdot}40,$

i.e., $25 = 76{\cdot}73 \,.\, \eta_m \,.\, p_m \,.\, D^2.$

Substituting $\eta_m = 0{\cdot}75$, and $p_m = 6{\cdot}9$ kg. cm.^{2} (98 lbs. per sq. inch),

then $25 = \text{about } 397\, D^2,\ D^2 = \frac{25}{397} = 0{\cdot}063 \text{ m.}^2,$

and $D = \text{about } 0{\cdot}25 \text{ m.}$

Thus, for 25 B.H.P. at 220 revolutions per minute, with a possible overload capacity of 20 per cent., the engine cylinder should have the following dimensions :—

$D = 250$ mm. and $S = 400$ mm. . . . (L.W.)

(2) *Four-cycle Diesel Engine* 50 *B.H.P. per cylinder.* $n = 190$. In this case use

$v = \text{about } 3{\cdot}20$ metres (10·5 feet) per second.

and from $S = \frac{30 \times 3{\cdot}20}{190}$

is obtained $S = \text{about } 0{\cdot}51$ metre,

and $N_e = 50 = 0{\cdot}872 \,.\, \eta_m . \, p_m \,.\, D^2 \,.\, 190 \times 0{\cdot}51,$

and $50 = \text{about } 84 \,.\, \eta_m \,.\, p_m \,.\, D^2.$

Substituting $\eta_m = \text{about } 0{\cdot}76$ and $p_m = \text{about } 6{\cdot}9$ kg. cm.^{2} (98 lbs. per square inch)

gives $D^2 = \text{about } 0{\cdot}114 \text{ m.}^2$ and $D = \text{about } 0{\cdot}34$ m.

This cylinder, therefore, has

$D = 340$ mm. and $S = 510$ mm. . . . (Sulzer)

(3) *Four-cycle Diesel Engine* 100 *B.H.P. per Cylinder.* $n = 165$.

$v = \text{about } 3{\cdot}70$ metres (12·1 feet) per second.

$S = 0{\cdot}675$ m.

$N_e = 100 = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, 165 \times 0{\cdot}675.$

By substituting $\eta_m = 0{\cdot}76$ and $p_m = 6{\cdot}7$ kgs. per cm.^{2} (95 lbs. per sq. inch)

is obtained $D = 0{\cdot}45$ m.,

and so $D = 450$ mm. and $S = 675$ mm. . . . (Tosi)

(4) *Four-cycle Diesel Engine* 150 *B.H.P. per Cylinder.* $n = 155$. For this relatively high power

v should be about 4 metres (13·1 feet) per second.

$$S = \frac{30v}{n} = \frac{30 \times 4}{155} = 0{\cdot}775 \text{ m.}$$

assuming $S = 780$ mm.,

and so $N_e = 150 = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, 155 \times 0{\cdot}78.$

Substituting $\eta_m = 0{\cdot}78$ and $p_m = 6{\cdot}7$ kgs. per cm.^{2} (95 lbs. per sq. inch)

$150 = 0{\cdot}872 \times 0{\cdot}78 \times 6{\cdot}7 \,.\, D^2 \,.\, 155 \times 0{\cdot}78$

$D^2 = 0{\cdot}272 \text{ m.}^2$ and $D = \text{about } 0{\cdot}520$ m.

then $D = 520$ mm. and $S = 780$ mm. . . . (M.A.N.)

(5) *Four-cycle High-speed Diesel Engine of* 40 *B.H.P. per Cylinder.* $n = 375$.

Putting $v = 3{\cdot}75$ metres (12·3 feet) per second
$S = 300$ mm. is obtained.

Whence $N_e = 40 = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, 375 \times 0{\cdot}30$.

By substituting $\eta_m = 0{\cdot}7$ and $p_m = 7$ kgs. cm.^{2} (100 lbs. per square inch)

$N_e = 40 = 0{\cdot}872 \times 0{\cdot}70 \times 7 \,.\, D^2 \,.\, 375 \times 0{\cdot}30,$

$D^2 = 0{\cdot}0832$ m.^{2}, $D = 0{\cdot}288$ m., say 0·29 m.

From this follows $D = 290$ mm. and $S = 300$ mm.

In this case it is to be noted that, in order to keep the piston speed within limits, the usual value of $\frac{S}{D}$ is departed from.

(6) *Two-cycle Engine of* 250 *B.H.P. per Cylinder at* 157 *Revolutions per Minute* (for the direct drive of an alternator with a periodicity of 42).

Putting $v =$ about 3·75 m. (12·3 feet) per sec.,

$$S = \frac{30 \times 3{\cdot}75}{157} = 0{\cdot}717 \text{ m.} = \text{about } 0{\cdot}720 \text{ m.}$$

$N_e = 250 = 1{\cdot}743 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, 157 \times 0{\cdot}72$.

Substituting $\eta_m = 0{\cdot}72$ and $p_m = 7$ kgs. cm.^{2} (100 lbs. per square inch),

$D^2 =$ about 0·25 m.^{2} and $D = 0{\cdot}50$ m.,

and so $D = 500$ mm., $S = 720$ mm. . . . . (Sulzer)

(7) *Two-cycle Two-cylinder Horizontal Engine, giving* 750 *B.H.P. at* 150 *Revolutions per Minute.*

With $v = 4$ metres (13·1 feet) per sec.

$$S = \frac{30 \times 4}{150} = 0{\cdot}80 \text{ metre,}$$

$N_e = 375 = 1{\cdot}743 \,.\, \eta_m \,.\, p_m \,.\, D^2 \times 150 \times 0{\cdot}80$.

Substituting $\eta_m =$ about 0·72 and $p_m = 6{\cdot}5$ kgs. per cm.^{2} (93 lbs. per sq. inch)

$D =$ about 0·42 m.

then $D = 420$ mm. and $S = 800$ mm. . . . (M.A.N.)

To calculate for a first approximation, a constant mean value may be assigned to the product of $\eta_m \,.\, p_m$, and this product included with the other constants in a new constant K, thus :—

$$N_e = K \,.\, D^2 \,.\, S \,.\, n,$$

in which the value K * in usual cases of land engines is

$K = 4{\cdot}5$ to 4·6 for four-cycle engines,

and $K = 8{\cdot}4$ to 8·8 for two-cycle engines.

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* This is the formula and the value of K already quoted in the note, p. 67.

For **marine engines** the values of p_m, η_m, n, and v to be substituted in the two formulæ,

$N_e = 0{\cdot}872 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, n \,.\, S$ for four-cycle engines,

and $N_e = 1{\cdot}743 \,.\, \eta_m \,.\, p_m \,.\, D^2 \,.\, n \,.\, S$ for those working on the two-cycle principle, which vary between wider limits than those given for land engines,

are :—

$v = \frac{2 \,.\, n \,.\, S}{60} = 3$ to 4 metres (9·85 to 13·1 feet) per second for small to large slow-speed engines;

3·50 to 5 or even up to 6 metres (11·5 to 16·4 or even 19·7 feet) per second for high and very high-speed engines, according to their power.

n = 90 to 150 for heavy engines of cargo boats;

150 to 250 for engines of the same type of small power;

300 to 500 in high-speed and light engines, such as those suitable for submarines;

400 to 600 in small engines for launches and the like.

η_m has the values already stated for land engines—*i.e.*, 0·75 to 0·80 for four-cycle slow-running engines; about 0·7 for high-speed four-cycle engines and for two-cycle engines; about 0·65 for high-speed two-cycle engines.

p_m = 6·5 to 7 kgs. cm.^{2} (92 to 100 lbs. per square inch), but certain types of light engines, which are run at their limit of power, have higher mean pressures than given, whilst those for cargo boats, intended for long non-stop runs, work at lower mean effective pressures.

The value of the product of $\eta_m \,.\, p_m$, which may be said almost to be characteristic of the engine, should be :—

5·6 to 5 for small to large cylinders of slow-running four-cycle engines.
4·8 to 4 for slow-running two-cycle engines.
5·5 to 4·5 for high-speed four-cycle engines.
4·5 to 4 for high-speed two-cycle engines.

The figures given above, it will be seen, show a wide variation. (They are collected from some thirty different engines.) In addition to being affected by the power and speed of revolution, this variation depends, firstly, on the fact that in some designs a greater margin for overload is allowed than in others; secondly, that in cargo boats and all ships having to make long voyages with absolute reliability, the engine is made large (*i.e.*, the specific power of the cylinder volume is kept low); and thirdly, because the power of an engine is frequently stated in round numbers, either above or below the actual figure, by a considerable percentage.

Engine makers sometimes use the same patterns in the foundry for engines of slightly different powers, varying the number of cylinders, the piston stroke or the revolutions, and this leads to differences in efficiency or of the mean effective pressure.

Some makers prefer the policy of allowing a margin, so that an engine developing, say, 600 B.H.P. on trial is called a 500 B.H.P. engine. Others,

for commercial considerations, construct engines without any margin of power, and, in consequence, their full power is only developed on trial, when the engine is running under the most favourable conditions.

In practice, it is generally known that the main characteristic of an engine—its power—is to some extent a matter of opinion, with marine engines

Maker of Engine.	Built for	Name of Ship.
LOW-SPEED FOUR-CYCLE ENGINES—		
Burmeister & Wain, Copenhagen,	East Asiatic Co.	"Selandia."
Lud. Nobel, Petrograd, . .	Nobel Bros.	"Robert Nobel."
Werkspoor, Amsterdam, . .	Dutch Indies Tank Ship Co.	"Vulcanus."
,, ,, . .	Sonda Islands Co.	"Sembilan."
LOW-SPEED TWO CYCLE ENGINES—		
Reiherstieg, Hamburg, . .	German-American Petroleum Co.	"Excelsior."
Fiat, Turin, ...	..	..
Tecklenborg, Bremerhaven, .	"Hansa," Bremen.	"Rolandseck."
Carels, Ghent, ...	..	..
Richardson, Westgarth, Middlesbrough and Carels.	Furness, Withy & Co.	"Eavestone."
Gebr. Sulzer, Winterthur, . .	Hamburg South America Line.	"Monte Penedo."
A. G. Weser, Bremen, & Junker,	Hamburg-America Line.	..
Franco Tosi, Legnano, . .	Italian Navy.	..
Gebr. Sulzer, Winterthur, .	Sansone Forli, Ravenna.	"Romagna."
HIGH-SPEED FOUR-CYCLE ENGINES—		
Lud. Nobel, Petrograd, . .	Russian Navy.	..
M.A.N., Augsburg, . . .	German Navy.	Submarine.
Sabathé, St. Etienne, . .	French Navy.	..
Act. Diesel, Stockholm, . .	..	..
Normand, Havre, . . .	French Navy.	..
M.A.N., Augsburg, . . .	..	..
Gebr. Körting, Hanover, . .	..	..
HIGH-SPEED TWO-CYCLE ENGINES—		
Gebr. Körting, Hanover, . .	German Navy.	Submarine.
M.A.N., Nürnberg, . . .	Russian Navy.	,,
Gebr. Sulzer, Winterthur, . .	Italian Navy.	,,
M.A.N., Nürnberg, . . .	..	..
Gebr. Sulzer, Winterthur, . .	Italian Navy.	Tug No. 19.
Fiat, Turin, ...	..	..
M.A.N., Nürnberg, . . .	..	..

even more than with land engines. The latter are of a more constant type, and in them certain dimensions are nowadays almost stereotyped.

In view of these considerations, the following table gives numerous examples of cylinder volumes and revolutions of marine engines, classified according to the speed of revolution and to the cycle of operation :—

N_e.	Number of Cylinders.	B.H.P. of each Cylinder.	D in mm.	S in mm.	n	v	$\eta_m \cdot p_m$
1,000	8	125	530	650	140	3·0	5·60
400	4	100	450	510	215	3·65	5·15
500	6	84	400	600	180	3·60	5·50
210	3	70	400	500	200	3·35	5·0
1,800	6	300	600	1,100	90	3·25	4·80
1,000	4	250	550	800	150	4·0	4·0
1,500	6	250	510	920	130	4·0	4·65
900	4	225	450	560	250	4·65	4·60
800	4	200	508	914	100	3·05	4·8
800	4	200	470	680	160	3·60	4·8
800	2 × 3	133	400	2 × 400	120	2 × 1·60	5·0
500	4	125	400	650	170	3·70	4·10
310	4	78	310	460	225	3·45	4·60
600	4	150	450	510	310	5·30	5·35
850	6	142	400	400	450	6	5·56
500	6	83	350	350	400	4·70	5·60
450	6	75	350	350	400	4·70	5·30
420	6	70	330	360	400	4·80	5·10
100	6	16·5	165	250	600	5·0	4·45
100	6	16·5	190	240	500	4·0	4·35
850	6	142	350	350	425	5·0	4·4
850	8	106	300	340	450	5·10	4·4
330	6	55	230	280	500	4·65	4·3
100	4	25	180	300	350	3·50	4·25
100	4	25	180	250	390	3·25	4·5
100	4	25	170	220	500	3·70	4·5
100	4	25	175	200	600	4·0	3·9

PART II.

CHAPTER VII.

BED PLATES, CRANK CASES, ENGINE FRAMING, CYLINDERS.

Bed Plates of Stationary Engines.—The bed plate is attached to the foundation or engine seating, contains the main bearings of the crank shaft, and provides for the attachment of the crank case or engine columns (Fig. 66). The bed plate is secured to the foundation by the usual holding-down bolts, and to the columns or crank case, by strong studs, which hold the carefully machined surfaces together. The bed plate and crank case are rarely cast in one piece, and then only for the very smallest powers.

Generally, the bed plate is of cast iron, though sometimes, with high-speed and light engines, it is of cast steel or bronze. It is cast in one piece,

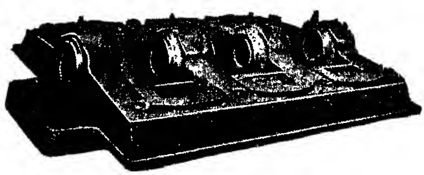

Fig. 66.—Bed Plate of small Stationary Engine.

even in the case of multi-cylinder engines. Only in units of six or eight cylinders of high power is the engine divided in two halves, between which the flywheel is sometimes placed. When the bed plate is in more than one piece, it is bolted together with fitted bolts, and a key is fitted for ensuring alignment. This costly practice is only adopted when the dimensions of the bed plate are such as to give rise to excessive difficulties in the foundry, when the available planing machines are not sufficiently large, or when the question of transport may present difficulties.

Figs. 67 and 68 show a bed plate for a single cylinder normal speed engine; the two main bearings have bushes of cast iron lined with white metal, and automatic lubrication by means of rings. One of these bearings appears to be longer than the other, owing to the bushes being divided by the helical gearing for driving the intermediate vertical shaft, which at its upper end serves to rotate the cam shaft. The gearing and the footstep bearing supporting the vertical shaft work in the same bath of oil as the main bearing lubricating rings (see Fig. 69).

Bed plates for multi-cylinder engines are similar to those shown in Figs. 67 and 68, and have one bearing divided for the spiral gearing, and as many more similar to that shown in section (Fig. 68) as there are cylinders. The oil reservoirs of the various main bearings are in communication with one another through a pipe *a* (Figs. 67 and 68), and also with a gauge

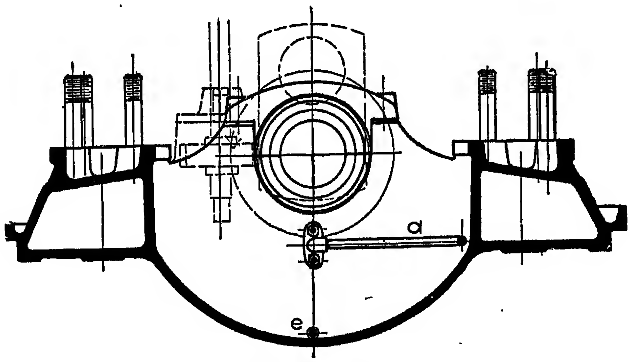

Fig. 67.—Cross-section of Bed Plate for Single-cylinder Engine.

glass *c*. Another pipe *e* connects the crank pits and serves also to drain off the oil. The oil which lubricates the bearings may, after being filtered, be used again with the addition of fresh oil. Any oil which falls from the piston is to a great extent burnt, and has a bad effect upon the lubricating oil.

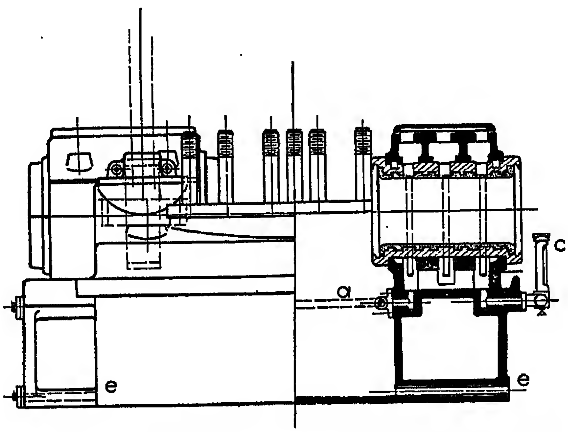

Fig. 68.—Bed Plate and Bearings for Single-cylinder Engine.

High-speed stationary engines have bed plates similar to those for high-speed marine engines, to be described later. They are lighter than those of slow-running engines, usually of stronger material, have forced lubrication for their bearings, and sometimes have water circulation under the bottom main bearing brass, and less frequently in the main bearing cap.

No rule can be given for the calculation of bed plates ; in each case after the section has been determined, the stresses can be calculated, and it can be judged whether the strength of the design is sufficient.* The stresses in this part should always be kept very low.

Bed Plates of Marine Engines—Thrust Blocks.—With marine engines the seating consists of the engine bearers, which are two stiff longitudinal beams of hollow section built up of plates and angle irons. To them the

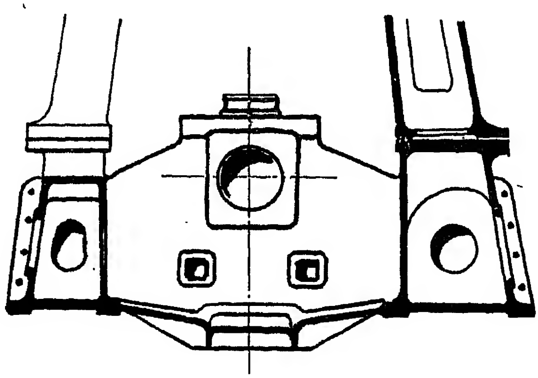

Fig. 70.—Cross-section of Bed Plate of Germaniawerft Mercantile Marine Engine.

bed plate is secured with numerous bolts, chocks of teak or other suitable material being frequently fitted between the bed plate and the bearers.

The bed plates of high-powered engines are divided into two or three sections jointed together with flanges ; they are of cast iron in heavy slow-running engines, of cast steel in other types, or even of high tensile bronze when extreme lightness is required.

Figs. 70 (Germaniawerft) and 71 (Junkers) represent the transverse

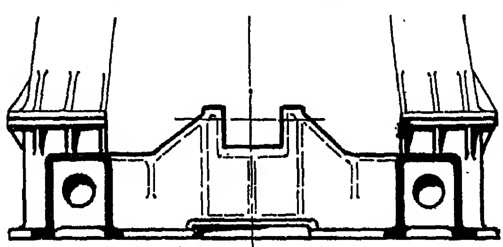

Fig. 71.—Cross-section of Bed Plate of Junkers Mercantile Marine Engine.

sections of two bed plates for mercantile marine engines. They consist of a framework of beams of hollow section intersected by the cross-girders, also of hollow rectangular section, carrying the main bearings. The crank pits are formed between the longitudinal and transverse beams.

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* Some examples of these calculations may be found in "The Design and Construction of Internal Combustion Engines," Güldner, p. 250, *et seq.*

Figs. 72 to 75 represent transverse sections of bed plates for high-speed engines.

Instead of the rectangular feet on the bed plate carrying the columns of the slow-running engine, this type has two machined faces running the full length of the bed plate, upon which the crank case, common to all the cylinders, is bolted. Figs. 72 and 73 show respectively the Sabathé and Nobel types of bed plates, and the latter has a number of feet for attaching it to the engine bearers, instead of the usual flange running fore and aft. Fig. 74 gives the deep and strong type adopted by the M.A.N., and the one shown in Fig. 75—the Sulzer type—is fitted with through bolts of forged steel, connecting the cylinders to the bed plate and thus removing the tension strains from the crank case or engine framing.

Almost all bed plates carry the main bearings immediately on each

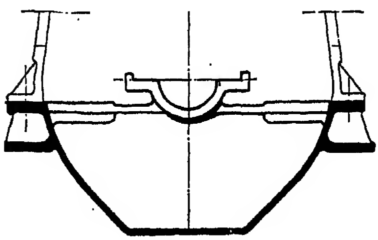


Fig. 72.

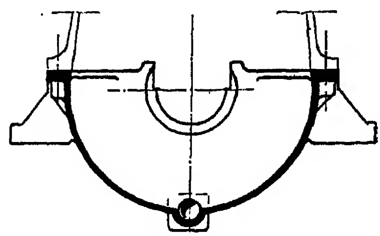


Fig. 73.

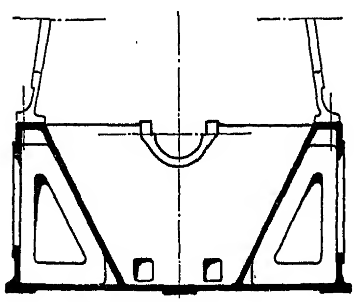


Fig. 74.

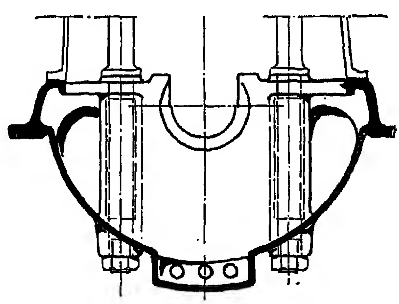


Fig. 75.

Figs. 72 to 75.—Typical Cross-sections of Bed Plates of High-speed Engines.

side of each crank, and these bearings have bushes of cast iron, bronze or cast steel, lined with white metal. In slow-running engines, lubrication is by drip feed or rings, whilst for high speeds forced lubrication is general. A system of forced lubrication gives the best results without necessitating a large consumption of lubricating oil, and is sometimes adopted even for the engines of cargo boats.

With such, however, it is necessary completely to enclose the cranks and connecting-rods in a casing, to prevent the oil being thrown out and wasted. With an enclosed engine the bearings cannot readily be felt whilst the engine is running. Forced lubrication, however, greatly reduces the probability of hot bearings.

The oil which escapes from the bearings subject to the lubricating oil pressure is collected in a sump at the bottom of the bed plate, where it is cooled by coils for water circulation in the sump (Fig. 75), or the sump itself may be externally water-cooled (Fig. 76).

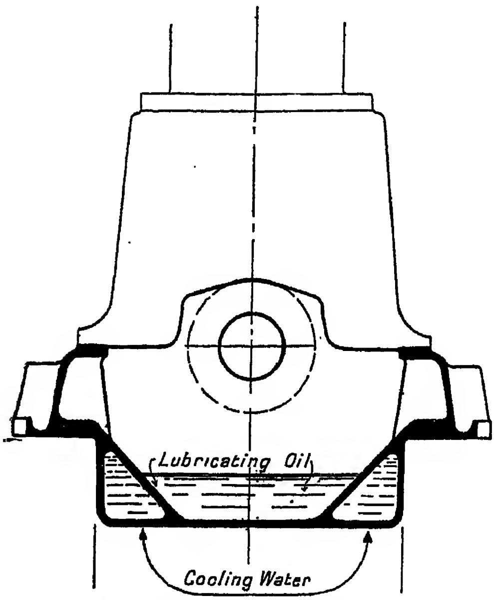
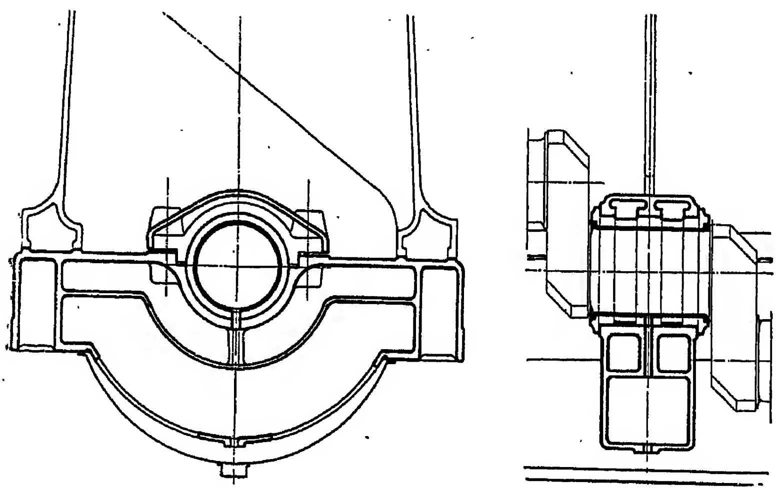


Fig. 76.—System of Cooling Lubricating Oil by External Reservoirs.

The main bearings themselves in high-speed and light engines are often cooled by water circulation in the bed plate under the lower bearing bush (Figs. 77 and 78, Fiat), and sometimes also through the bearing cap.

Thrust Blocks.—In all marine engines, provision must be made to take the axial thrust of the propeller shaft abaft the crank shaft of the engine.

The propeller in revolving forces the water astern, and by reaction tends to thrust the shafting forward through its bearings. This



Figs. 77 and 78.—Fiat System of Cooling Main Bearings by Water Circulation in Bed Plate.

axial load, to which the shafting is subjected, is additional to the torsion, and necessitates a thrust block abaft the crank shaft.

The thrust block may be incorporated with the bed plate, in which case the aftermost bearing is lengthened and fitted with thrust collars (Figs. 79 and 80, Sulzer).

Generally it is independent and separately secured to the ship's framing (Fig. 81).

With the first construction, sometimes adopted for small engines, the axial thrust is taken off the cranks, but the whole engine tends to slide forward on its seating, whilst with the second, preferable in high-power engines, no thrust is transmitted to any part of the engine framing.

For small powers Penn bearings, with fixed thrust rings, may be used; for large engines the Maudslay system, with adjustable shoes, is generally adopted, in which the thrust collars are not covered, and are always accessible to the hand when running (Fig. 81).

The thrust block ought only to be subject to the axial thrust, and not to the weight of the shaft. For this reason it is always placed close to the after crank-shaft bearing or a plummer block; or it may be cast in one piece with one or two plummer blocks.

The Framing of Stationary Engines.—The casting carrying the cylinders is secured to the bed plate with bolts or studs; in normal-speed land engines the cylinder and frame are almost always cast in one piece, with a separate liner, free to expand longitudinally (Figs. 82 to 86).

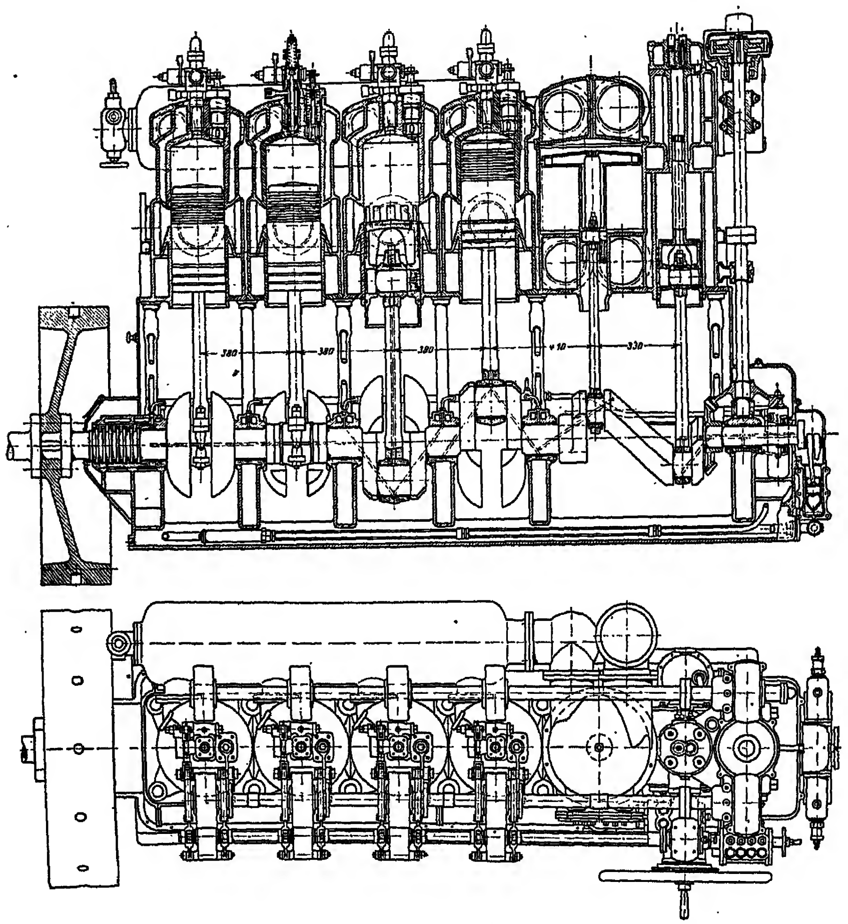
The joint between these castings and the bed plate should be carefully machined and then hand-scraped during erection, to ensure that the cylinders are truly perpendicular to the crank shaft and that the clearance space between piston and cylinder head is correct.

A dowel pin screwed into the bed plate to fit exactly into the hole *b* (Fig. 85) guarantees that the centre line of the frame shall coincide with that of the crank; this makes fitted bolts unnecessary.

The legs of the frame may be of hollow section (Fig. 87) or open with longitudinal ribs (*a*, Figs. 84 and 86), either one in the middle or one on each side. If the compressor is mounted at the back of the cylinder, an aperture is provided for the balance lever, driving the compressor through links from the main engine connecting-rod (Fig. 87). In some designs two apertures are provided in the legs similar to that for the balance lever of the compressor, to facilitate the examination and refit of the top end of the connecting-rod.

Holes (*c* in Figs. 85 and 86), fitted with bronze bushes, serve for the two bearings of the fulcrum shaft of the indicator driving lever, actuated by links and levers from a reciprocating part of the engine. Generally the small pump for lubricating the piston and the top end bearing is driven from the same fulcrum shaft as the indicator lever.

The lubrication of the piston is effected by forcing oil through a ring of copper piping, from which it enters the cylinder through four nipples which are screwed into the latter with packed glands to prevent leakage of water where they pass through the jacket (Fig. 149, p. 114). The nipples may be disposed at 90° to one another, and 45° to the axis of the shaft, or, since the working surfaces of the cylinder walls are those at the front and back, instead of being equidistant from one another, they may be closer to the transverse centre line of the cylinder. The open ends of the nipples should



Figs. 79 and 80.—Thrust Block incorporated in Bed Plate Casting (Sulzer Engine).

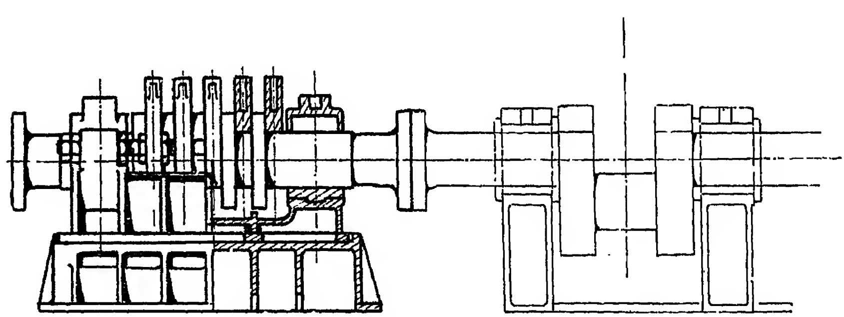


Fig. 81.—Example of separate Thrust Block.

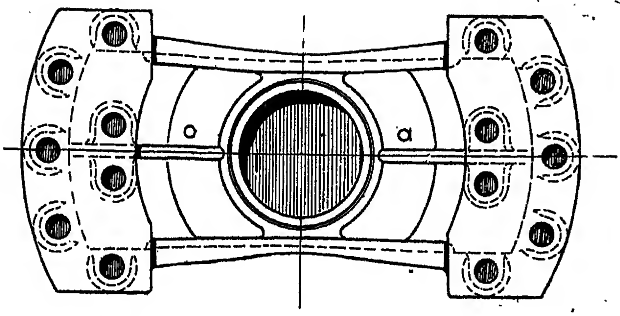
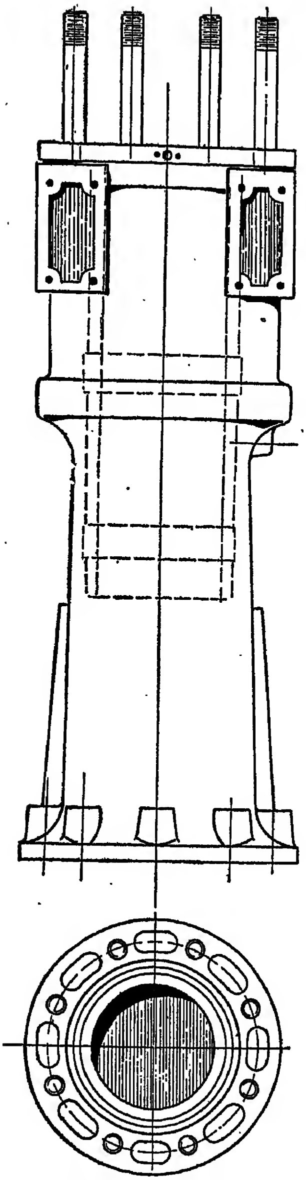


Fig. 84.



Figs. 82 and 83.

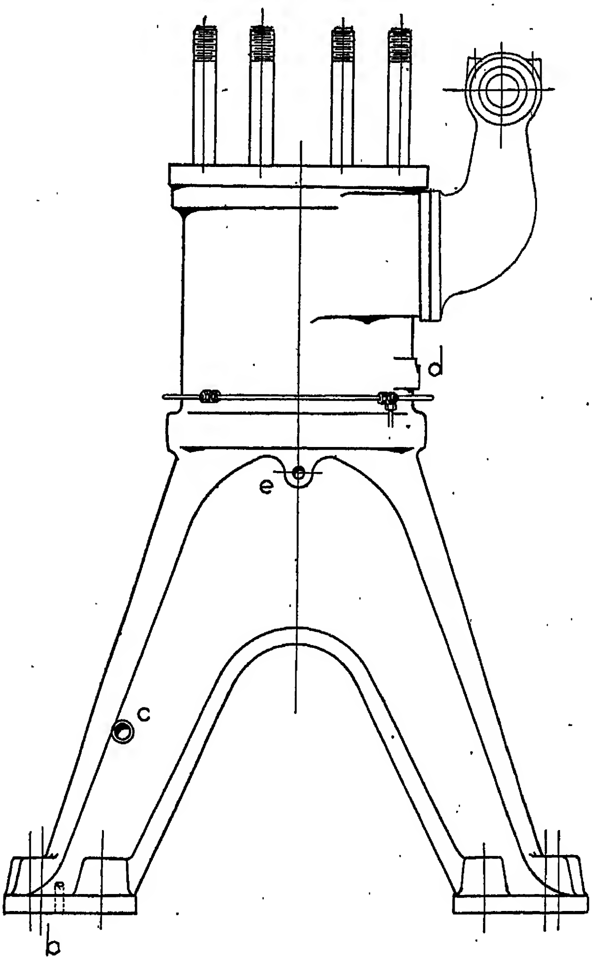


Fig. 85.

Figs. 82 to 85.—Example of Framing of Stationary Vertical Engine.

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be covered by the piston at all points of the stroke, and be placed near the position occupied by the top spring ring when the piston is at the bottom of its stroke.

In two-cycle engines these oil nipples are below the exhaust ports, although in some large units, for greater precaution, two series are provided, one immediately above, and the other immediately below, the exhaust ports.

The oil for the gudgeon pin enters through the hole *e* (Fig. 85), and is

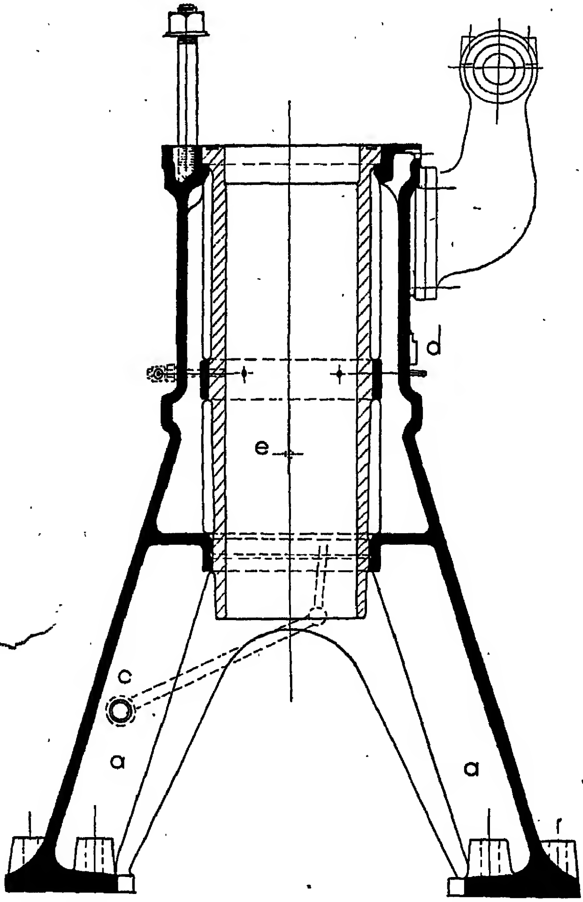


Fig. 86.—Hollow-Section Framing of Stationary Engine.

collected by a groove in the piston, as described later when dealing with this part.

The bracket *d* (Figs. 85 and 86) serves as a support for the platform, which extends along the whole front, and sometimes along the back, of the engine.

When the compressor is placed on the supporting bracket this is fitted

with a shoulder or with a dowel pin, to fix its position with the exactitude required by the small cylinder clearance spaces. The position of the brackets supporting the distributing or cam shaft, usually bolted to the frame, is also fixed in a similar manner. For simplicity of design some constructors carry these supporting brackets on the cylinder head, although this practice leads to further complications when dismantling (at all times rather a troublesome operation on account of the large number of pipes, etc., attached to the head). The removal of the head, when the bearing brackets are attached thereto, necessitates the taking away of the helical gearing and the cam

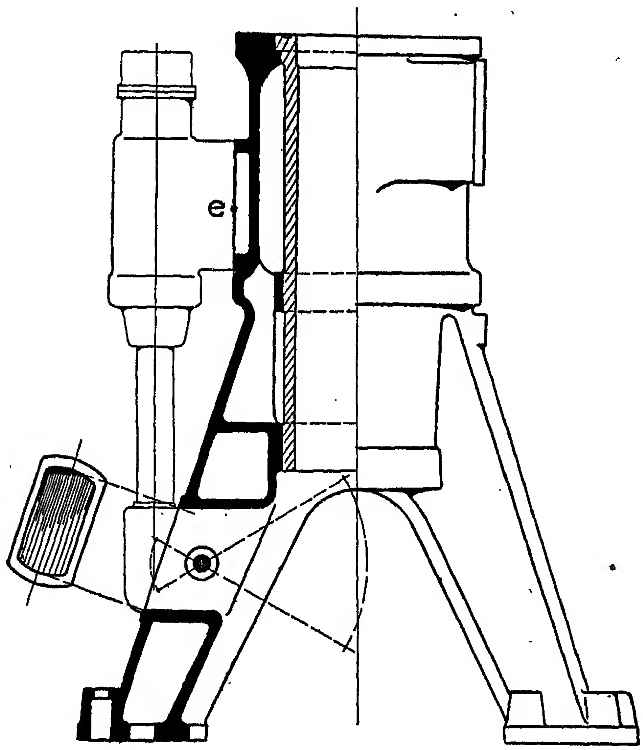


Fig. 87.—Framing for Stationary Engine with Compressor worked by Levers from Connecting-rod.

shaft, entailing a great deal of unnecessary work, especially when, in a multi-cylinder engine, it is desired only to remove one head.

The upper part of the frame is made considerably thicker than the remainder, to take the large studs for securing the cylinder head.

With a solid circumferential flange which eliminates the cooling effect of the water-jacket, the combustion chamber is kept hotter. In designs where the flange is not solid, as in Figs. 83 and 86, the lightening pockets open from the top, and they are not accessible for water, thus having the same effect of keeping the combustion chamber hotter.

The circulating water leaving the cylinder jacket generally passes through a short pipe to the cylinder head, so that the latter receives the water already slightly warmed * (Fig. 86).

Fig. 88 represents the frame for a two-cycle engine with valve scavenging, showing the annular exhaust passage cast with double walls for the cooling water-jacket.

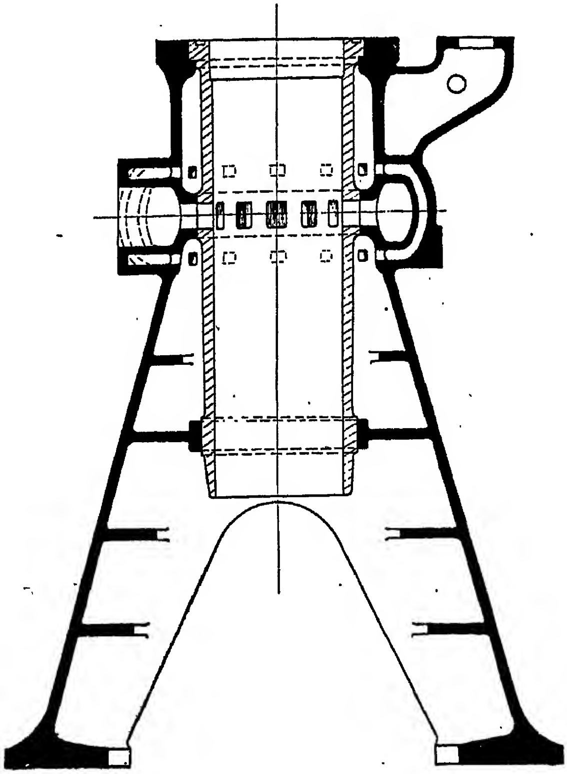


Fig. 88.—Frame and Cylinder of Two-cycle Vertical Stationary Diesel Engine with Valve Scavenging.

* In gas engines the head and the cylinder generally have two independent circulating water leads, but when, as is sometimes the case, there is only one lead, the water enters the head and passes to the cylinder jacket—*i.e.*, in the opposite direction to that which has been described for Diesel engines.

The reason for this is the necessity of keeping the combustion chamber of a gas engine as cold as possible, in order to be able to obtain a very high compression without danger of pre-ignition. In Diesel engines this danger does not exist. On the contrary, it is necessary that, at the end of the compression stroke of the Diesel cycle, the temperature of the air in the cylinder should be sufficiently high to insure the spontaneous ignition of the fuel.

And so, if it be assumed that for both types of engines the most suitable temperature for the cooling water in the cylinder jacket is the same, with Diesel engines the use of water of a higher temperature in the head gives an economy of water as compared with gas engines, even neglecting the water for the gas generator and scrubber.

The exhaust bars have a hole drilled or cored through them for water cooling—not shown in the diagram.

High-speed engines usually have light crank cases (in one piece even for multi-cylinder engines), to which the cylinder and generally also the compressor are bolted. This type of framing has been recently adopted for land engines (G.M.A., Neederlandsche, etc.), as has also the type in

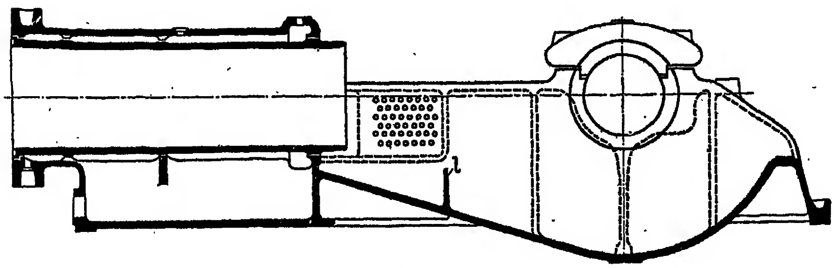


Fig. 89.

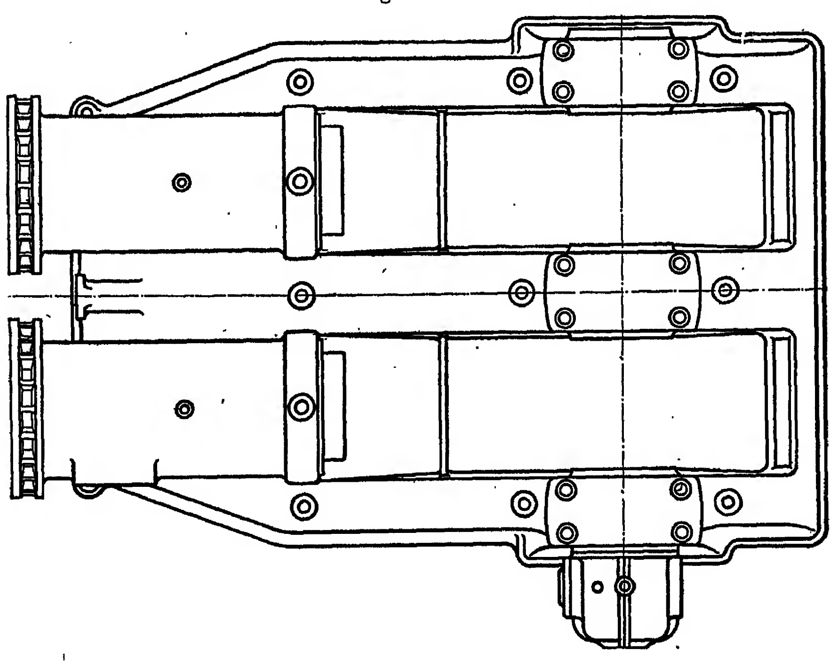


Fig. 90.

Figs. 89 and 90.—Bed Plate, Framing, and Cylinders of Horizontal Four-cycle Twin-cylinder M.A.N. Engine.

which the framing is cast solid with the cylinder-jacket (G.M.A., Sabathé, etc.). This latter type will be described when dealing with marine engines.

In the case of horizontal engines, the bed plate and the frame are often cast in one piece.

Figs. 89, 90, and 91 show the bed plates of two M.A.N. engines of this type, the one working on the four- and the other on the two-stroke cycle, both being twin-cylinder engines.

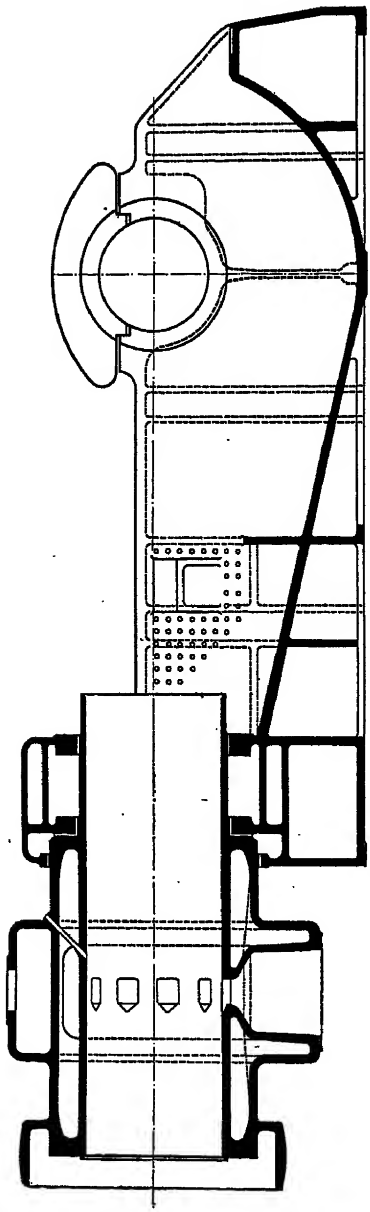


Fig. 91.—Bed Plate, Framing and Cylinder of Horizontal Two-cycle M.A.N. Engine.

The frames are similar to those for gas engines, excepting for their greater strength and the brackets and facings for the various parts peculiar to the Diesel engine—*e.g.*, air compressor, fuel injection pump, and, in two-cycle engines, the scavenging pump.

The cylinder lubrication is effected through a single nipple placed on the top centre line of the cylinder, where it is always covered by the piston. The oil tends to run down the walls of the cylinder, and is spread by the piston over the whole of the working surfaces.

The cooling water enters at the bottom close to the cylinder head, and is discharged at the outer end of the jacket at the top.

In large single- and double-acting engines the cylinder and its liner are sometimes cast in one piece separate from the frame.

The holes which are seen on the legs of the frame serve for the air suction, and the hollow interior acts as an air chamber and silencer.

The diaphragm (*l* in Fig. 89) prevents the dirty oil which drains from the piston from mixing with that flowing from the main and crank-pin bearings.

Framing of Marine Engines.—In marine Diesel engines examples are found of independent frames for each cylinder, and of crank cases in one or more pieces for the whole engine.

The crank case form of framing is specially suitable for high-speed engines, and may be of cast iron, cast steel, or sometimes of high tensile bronze. Generally, the cylinders and the compressor are attached to it by flanges,

with machined spigots for insuring their alignment, but examples are to be found where the dimensions of the engine or the capacity of the foundry permit of the cylinder jackets being cast in one piece with the crank case. The latter is strengthened by transverse ribs of rectangular or T section (Fig. 92), or by other methods of reinforcement according to the requirements of the particular case, and more especially to the form of cylinder and the method of its attachment.

Large inspection doors, opposite each crank, permit of the examination of the moving parts,* and of the expeditious carrying out of the small repairs and adjustments to the main bearing and crank-pin brasses.

To make possible the removal of the crank shaft without dismantling the whole engine, and to facilitate the work of refitting and examination, the crank case may have one side constructed so as to be readily removable, as shown in Fig. 93, representing a Sabathé engine. A similar construction is adopted for some engines built by the M.A.N. Augsberg, Nobel, Kolomna, etc.; the torpedo boat engine type of framing is sometimes met with in which

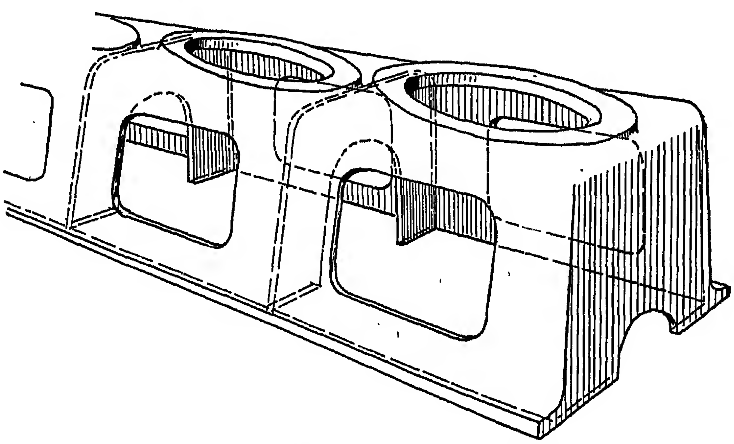


Fig. 92.—Crank Case Form of Marine Engine Framing with Transverse Ribs.

the cylinders are connected to the bed plate by forged steel columns, stiffened with diagonal stays (Sulzer, Figs. 79 and 80, Polar four-cycle, etc.).

Large engines are very similar, as regards the type of framing, to marine steam engines in which every cylinder is carried by two or three columns of cast iron or cast steel, which in turn carry the guides for the crosshead. This system is now almost universally adopted for Diesel engines of large powers (Fig. 94, Carels, Reiherstieg).

Figs. 95 to 99 (Plate V.) show the engine framing of the M.V. "Monte Penedo" (Sulzer). Through bolts of forged steel connect the bed plate to the cylinder heads. The columns and the cylinder itself are in this way relieved

* If the engine is fitted with forced lubrication, the doors cannot be opened while the engine is running, owing to the large quantity of oil which would be thrown out by the moving parts.

of all tension stresses consequent upon the piston loads, and serve only to absorb the lateral forces and to give stiffness to the engine. In way of the through bolts (which in this case number $2n + 2$, where n is the number of the cylinders) the cylinder heads and the columns are bolted together with flanges, so that the whole engine forms a rigid and compact structure.

In some four-cycle marine engines, frames identical in type with those for stationary engines are adopted, as illustrated in Figs. 82 to 87 (Nobel, Kolomna, etc.).

Cylinders.—In the case of both stationary and marine engines, the cylinders are almost always cast separate from the frames. They consist of three parts, the cylinder, cylinder liner and cylinder head. In some cases the

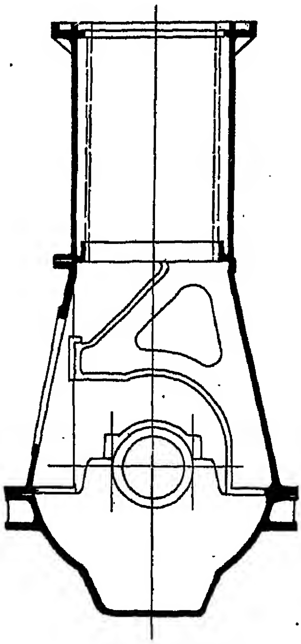


Fig. 93.—Sabathé Marine Engine Framing.

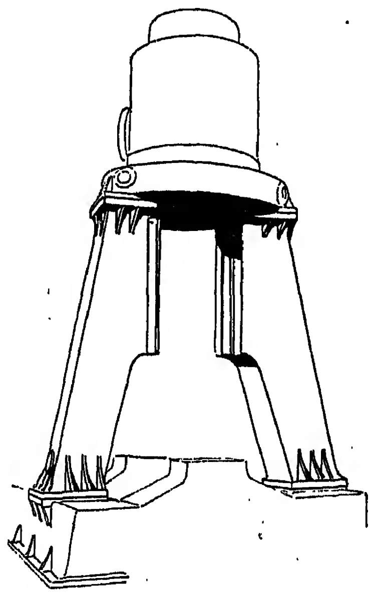


Fig. 94.—Carels Marine Engine Framing.

cylinder and liner are in one piece, with the intervening space serving for a water-jacket.

In small marine engines the cylinder, liner and head are cast in a single piece (Sulzer, Figs. 100 and 101, Fiat, M.A.N., Kind, etc.). In those of medium or large power the three parts are generally separate, the liner is pressed into the cylinder, and is free to expand longitudinally.* This construction has the advantage of freeing the liner from the axial stresses due to the pressure on the head. This is particularly important with two-cycle engines, where the strength of the liner is greatly reduced in way of the exhaust ports.

* The construction of the Germaniawerft shown in Fig. 105 (p. 91) is interesting. The head and the cylinder liner are cast in one piece, but are separate from the cylinder.

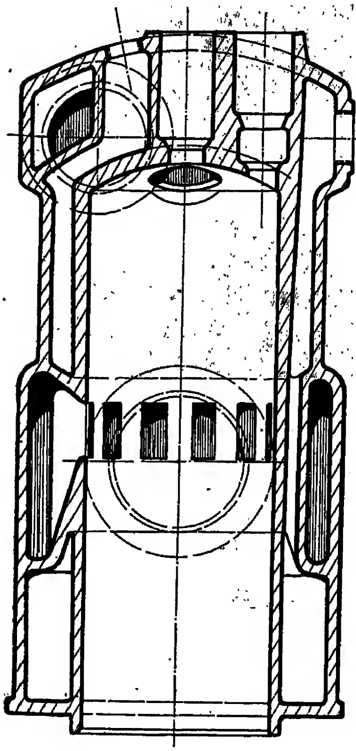


Fig. 100.

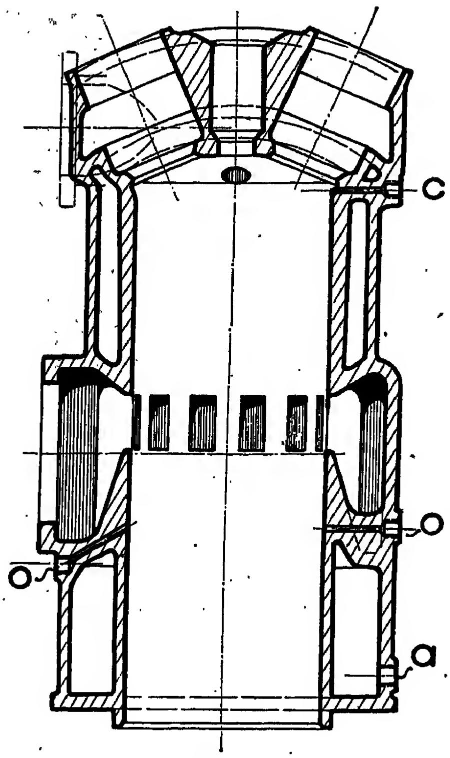


Fig. 101.

o = Piston lubrication. a = Entry of cooling water. c = Indicator cock.

TYPES OF CYLINDERS, CAST IN ONE PIECE, FOR SMALL MARINE ENGINES.

Generally the tension load is taken through the cylinder and the radial stress by the cylinder liner. In some designs the tension stresses are also removed from the cylinder by attaching the head to the frame direct by long bolts.

The liner is almost always of hard close-grained cast iron, cast with a large header. The jacket is of cast iron, or sometimes, in the case of light engines, of cast steel.

Fig. 102 represents the section of a liner for a four-cycle, and Fig. 104 that of a two-cycle engine, in both of which designs the liner is free to expand longitudinally in the cylinder. At the top the flange of the liner is shown resting in the cylinder, and this part ought to be carefully designed, in view of the heavy stresses imposed upon it by the spigot jointing ring C (Fig. 102), which insures tightness between the liner and the head. The groove for this joint may be machined entirely in the flange of the liner, or half in this and half in the cylinder (Fig. 103), obtaining by this second method a perfectly tight joint for the cooling water as well as for the cylinder gases. The bottom

of the liner is made a working fit in the cylinder to permit of free expansion. The water-tight joint between the liner and the cylinder at the bottom end is made by one or other of the methods in use for gas engines, sometimes by means of a gland, or by a rubber ring compressed by the liner into a groove *d* machined in that part of the cylinder which serves as a guide for the liner (Fig. 102). In addition to the top and bottom joints between

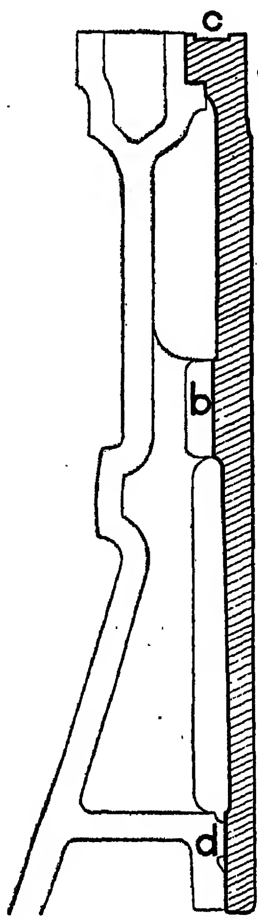


Fig. 102.—Section of Liner for Four-cycle Engine, showing Method of Jointing and Staying.

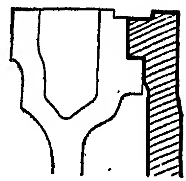


Fig. 103.—Alternative System for Spigot Jointing Ring *c* to ensure Tightness at Top Connection of Cylinder and Liner.

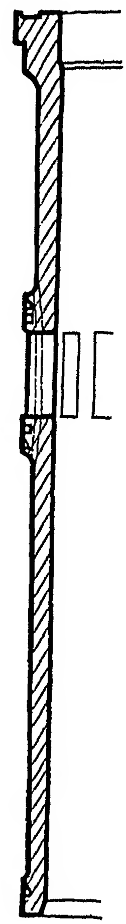


Fig. 104. Section of Liner for Two-cycle Engine.

the liner and the cylinder, to assist the liner in resisting the transverse thrust due to the obliquity of the connecting-rod, an intermediate point of support *b* is generally introduced. In two-cycle engines, this extra support is given in way of the exhaust ports.

The exhaust ports, as already stated, have a length of about 20 per cent. of the stroke, and are separated by bars, preferably of ample section, with a hole of as large diameter as possible for water circulation.

The arrangement for the cooling of these bars requires the very closest attention, owing to the fact that they are exposed each stroke to the flame of the burning gases.

The top of the liner should be bored out to an internal diameter slightly larger than that of the piston down to the position occupied by the centre line of the top piston ring, when the crank is at the top dead centre. This bell-mouthing *a* (Fig. 102) prevents the formation of a ridge on the liner wall by the top ring at the top

point of its travel. The piston would knock against this ridge at every revolution. This bell-mouthing also facilitates the entry of the piston and its rings into the liner.

The thickness of the cylinder liner is generally reduced towards the bottom. That of the upper part subjected to the highest pressures may be calculated from the formula (Bach)—

$$\delta = \frac{D}{2}\left\{\sqrt{\frac{K + 0{\cdot}4p}{K - 1{\cdot}3p}} - 1\right\}.$$

in which D is the diameter of the cylinder, K the stress in the material in kgs. per sq. cm., and p the maximum pressure of the indicator diagram.

Putting K = 250, and p = 30 atmospheres,

$$\delta = \frac{D}{2}\left\{\sqrt{\frac{250 + 0{\cdot}4 \times 30}{250 - 1{\cdot}3 \times 30}} - 1\right\} = 0{\cdot}055D.$$

To the value of δ obtained as above a constant of 15 mm. for small to 10 mm. for large engines is usually added to allow for the necessary regrinding, when through time the cylinder liner has worn oval. This constant introduces a factor of safety, and makes it unnecessary to use more than average care in assigning a value to p.

The length of the liner is determined by the stroke and the piston length. The piston may be allowed to overrun the liner at the bottom dead centre for about one-fifth of the piston length where no crosshead is fitted, whilst with a crosshead the piston may project nearly as far as the lowest piston ring.

When the liner has to resist the bending moment produced by the horizontal component N (Fig. 140, p. 108) of the pressure load due to the obliquity of the connecting-rod, it is desirable to check the unit stress in the liner and the deflection which this moment will cause.

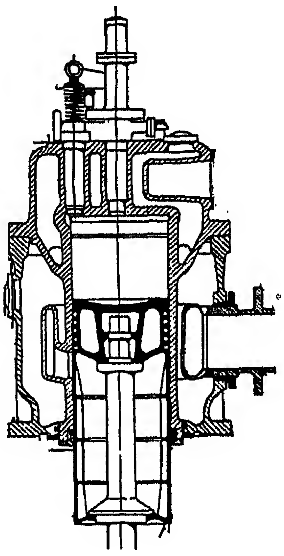


Fig. 105.—Example of Two-cycle Engine having Cylinder Liner and Head Cast in One Piece with Separate Jacket.

It is safe to assume, in the calculation, that the load, instead of being distributed over the cylindrical part of the piston, is concentrated at a section corresponding to the axis of the gudgeon pin.

It is sufficient for the purposes of calculation to take the position of the piston in which the bending moment is a maximum. This position is readily found by trial, or graphically from the diagram of forces N (p. 108, Figs. 140 and 141).

Should the unsupported length of the liner be such that the deflection, due to the force N, requires an increase of the section modulus, it is generally preferable to provide for this by introducing longitudinal ribs rather than by increasing the wall thickness, and so diminishing the cooling effect of the water.

CHAPTER VIII.

CRANK SHAFTS, CONNECTING-RODS, PISTONS.

Crank Shafts.—The crank shafts are made of Siemens-Martin steel with an ultimate tensile strength of 50 to 55 kgs. per square cm. (32 to 35 tons per square inch) and an elongation of over 20 per cent.

Land and high-speed marine engines, even with four or six cylinders, excepting those of high power, generally have their crank shafts in one

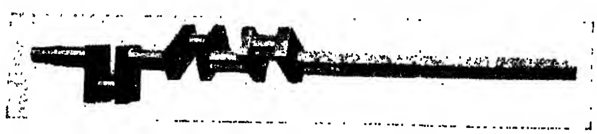
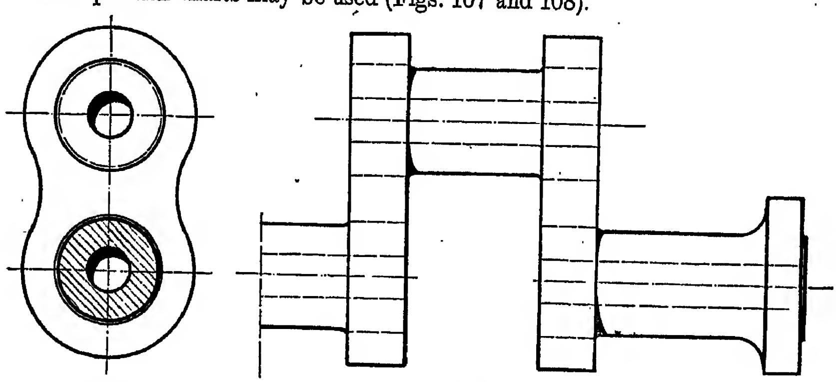


Fig. 106.—Multiple Crank Shaft in One Piece.

piece (Fig. 106); but in the case of cargo-boat engines the crank shaft is generally divided into sections of two or three cranks each.

Owing to the high maximum cylinder pressures, crank shafts forged at a high temperature from a bar should not be used even for very small oil engines. With high-power marine engines having a large stroke bore ratio built-up crank shafts may be used (Figs. 107 and 108).

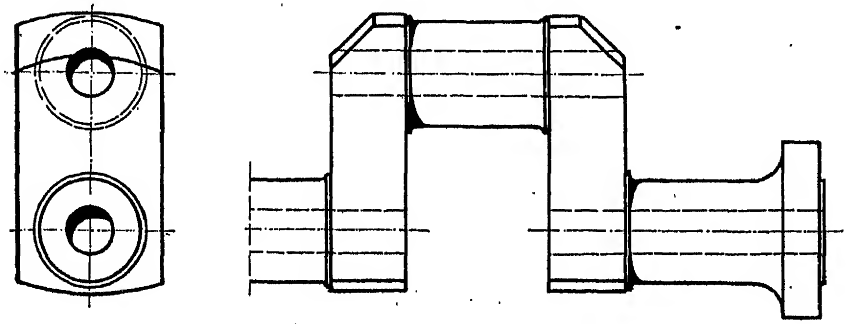


Figs. 107 and 108.—Built-up Crank Shaft for High-power Marine Engines.

Generally, for all classes of Diesel engines, the crank shafts are solid forged (Figs. 109 and 110), in which case the shaft is forged under a hydraulic press, leaving rectangular blocks where the cranks are to be formed. Fig. 111 shows the outline of the rough forging, and from these blocks by means of drilling and slotting machines a smaller block (shown in dotted lines in the

figure) is cut, so that from the remaining part the pin and the arms of the crank may be turned and shaped. The fine lines in the figure show the finished crank.

The forging of these shafts, especially those for high-power engines, requires plant of such size that only the largest forges can undertake the work.



Figs. 109 and 110.—Solid Forged Crank Shaft for Diesel Marine Engines.

In Italy the crank shafts are imported from abroad, and are generally supplied rough-machined.

To detect flaws in the metal, and for lightness, the crank shafts and pins are often bored hollow, to a diameter between 0·4 and 0·6 that of the shaft or pin. As a rule, account is not taken of this reduction in strength when making calculations for the strength of the shaft.

It is difficult sufficiently to support long shafts during the operations of turning (especially when turning the crank pins with the shaft revolving eccentrically), as the unsupported weight may cause appreciable bending, and the pressure of the tool is apt to give rise to vibration. On this account, special crank-pin machines, in which the shaft remains fixed, whilst the tool revolves around the pin, are often adopted.

Fig. 112 represents a finished crank shaft of a two-cylinder stationary Diesel engine.

The cranks are at 360° to one another. The small crank *c* drives the compressor; *a*, *a* and a_1, are the journals of the main bearings, the last of which contains the gear wheel for driving the valve gear; *e* is the tail end bearing journal.

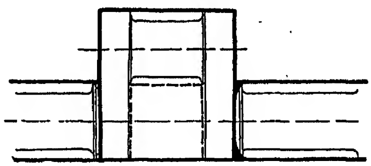


Fig. 111.—Rough Forging for Solid Crank Shaft.

The flywheel is carried by the enlarged part *g*, in which two keyways are cut.

Centrifugal lubrication is provided for the crank-pin bearing. The centrifugal lubricators are of bronze or malleable cast iron, attached to all the cranks, including that of the compressor. This latter is frequently separate from the remainder of the shaft, and secured to it with sunk-headed screws.

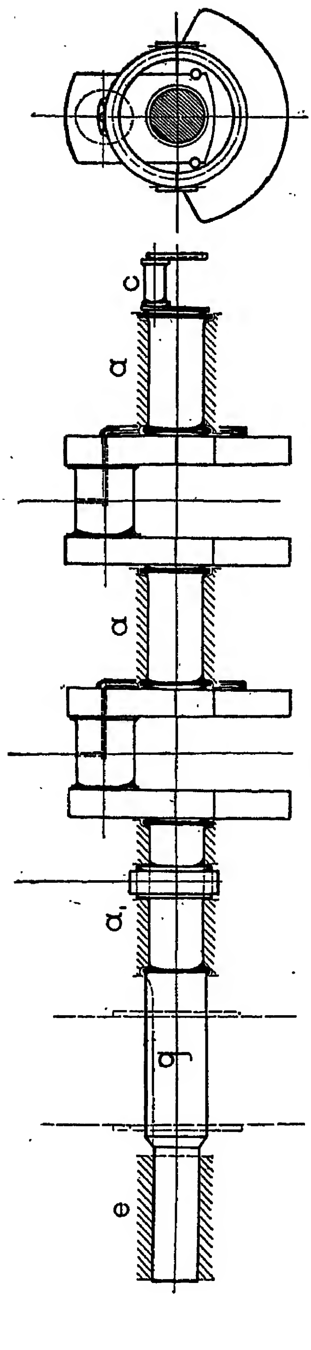


Fig. 112.—Crank Shaft of a Two-cylinder Stationary Diesel Engine.

The angles between the cranks are designed to give equal periods of time between the combustion strokes of the various cylinders, and to balance, as far as possible, the inertia forces of the reciprocating and rotating masses.

Thus, with *four-cycle engines*, the cranks are at 360° if there are two cylinders, at 240° with three, and 180° and 120° respectively for engines with four and six

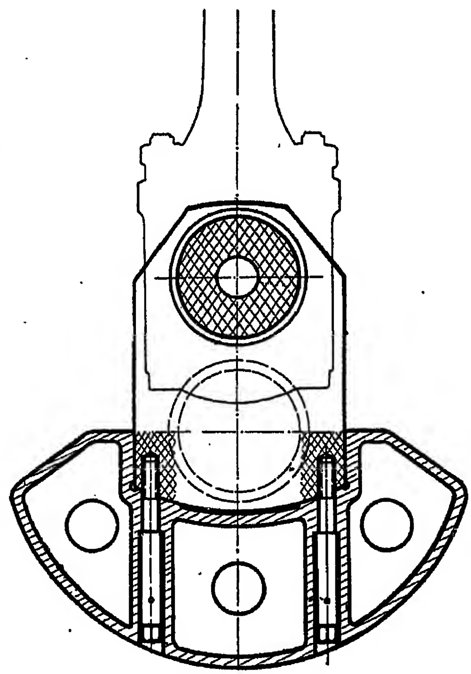


Fig. 113.—Counterbalanced Crank of Two-cylinder Engine.

cylinders. The cranks must be counterbalanced with two-cylinder engines (Fig. 113),* to prevent excessive vibration in the foundations, or in the hull in the case of marine engines. It is impossible to obtain

* The balance weight shown in section (Fig. 113) is usually of cast iron, hollow, and filled with lead. The attachment to the crank may also be made by means of keys dove-tailed into the two pieces.

perfect equilibrium of the reciprocating masses,* and for this reason the two-cylinder arrangement, which is common in land practice, is very little used for marine purposes, since in addition to its other disadvantages, it does not satisfy the conditions of smooth running.

The cranks of three-cylinder engines are not fitted with balance weights, except for high speeds, although in this case, also, the forces are not all in equilibrium, and produce a rocking couple.

In four- or six-cylinder engines the balance between the cylinders is practically perfect, and the designer need only provide for the inertia of the relatively small moving parts of the compressor.

In *two-cycle engines* with two cylinders the cranks are at 180° to one another; with three cylinders at 120°, and in four and six cylinders at 90° and 120° respectively, and only with this last is the balance perfect. For the others, especially in high-speed engines, the rotating masses should preferably be balanced. When the scavenging pump is driven from a separate crank on the engine crank shaft, its masses must also be balanced.

The compressor is often arranged parallel with the scavenging pump, and driven by a balance lever from the crank of the latter, so that the acceleration forces of the reciprocating masses of the two compensate for one another at least in part.

Lubrication of the bearings, as stated in the preceding chapter, is effected by means of drip, ring, or, in high-speed engines, forced lubrication. That of the crank-pin bearing, if not under forced lubrication, is provided for by centrifugal lubricators of bronze or malleable cast iron fitted to the crank webs.

Forced lubrication is carried out by a plunger or rotary pump, delivering the oil into the cap of each main bearing; some of this oil, spreading itself over the journal, leaks out to the save-all of the bed plate, whilst the remainder passes to the crank pin through a hole drilled in the shaft (Fig. 114). Again, part lubricates the crank pin and escapes, whilst the remainder is delivered to the gudgeon pin through a hole drilled in the connecting-rod.

The oil which leaks from the main bearing, crank-pin, and gudgeon-pin journals is all collected in the save-all of the bed plate, where, as seen in the preceding chapter (Figs. 75 and 76, on pp. 77 and 78), it is cooled before returning into circulation.

Sometimes, especially in high-speed marine engines, the pressure circuit of the oil extends to the piston head.

In many cases, as explained later, it is desirable to cool this part efficiently, subjected as it is to high pressures and very high temperatures.

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* The centrifugal force of the rotating masses (crank webs, crank pin, bottom end of connecting-rod, and half to one-third of the connecting-rod proper, are easily balanced by weights on the cranks. If it is wished to balance the inertia of the reciprocating masses (the piston, two-thirds to half of the connecting-rod, piston-rod and cross-head, if fitted) by means of rotating weights, the horizontal components of the centrifugal forces of these weights tend to set up horizontal vibration in the engine in a direction perpendicular to the axis of the shaft. One disturbing force is thus eliminated, and another introduced. Usually, balancing is limited in vertical engines to compensating the centrifugal forces of the rotating masses, both for the foregoing reasons and for the obvious constructive difficulty inherent in the application of heavy weights to a crank shaft.

Some constructors prefer oil to water cooling (M.A.N., Fiat, etc.), in which case the lubricating oil, after having reached the gudgeon-pin bearing, is led, through passages in the gudgeon pin and in the walls of the piston, to an enclosed space in the piston crown, whence it is discharged back into the save-all of the bed plate.

Diesel engine crank-shaft calculations are similar to those for all piston engines, and are based on the diagram of tangential forces. The diameter of the crank pin is made equal to that of the crank-shaft journals. Calculations give different values, and the larger of the two may be adopted for both.

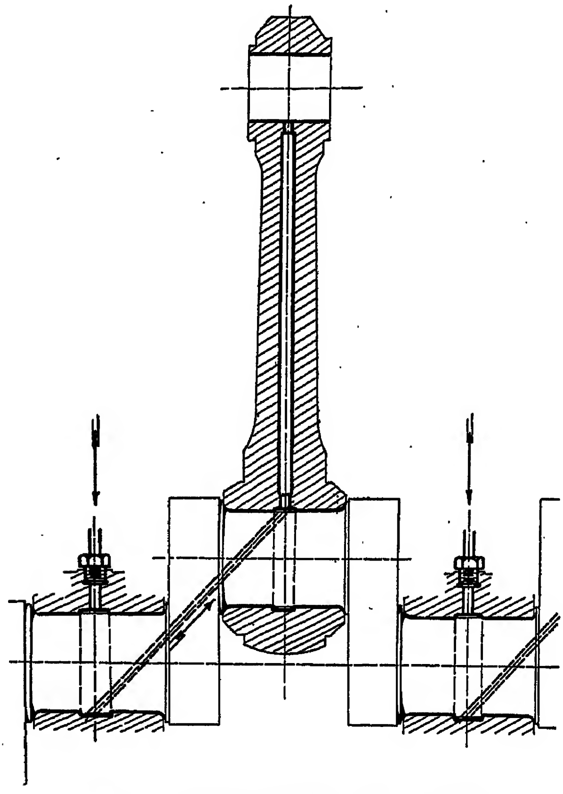


Fig. 114.—Forced Lubrication Arrangements.

The maximum stress allowable is 10 or 11 kgs. per square mm. (14,000 to 16,000 lbs. per square inch), taken out on a piston pressure of 45·to 50 atmospheres (640 to 700 lbs. per square inch), which is met with during the period of starting under compressed air.

Having determined the leading dimensions of the crank shaft for a single-cylinder engine, those for a two-cylinder engine of the same piston diameter and stroke may be similar, since the maximum stresses at any instant do not greatly exceed those with a single cylinder.

With four-cycle engines of three or four cylinders, the crank shaft dimensions are the same as those for a single-cylinder engine of the same piston diameter and stroke.

Since the calculations for crank shafts are of the same type for all reciprocating engines, it is not deemed necessary to treat this subject further here, and the reader is referred to the classical works of Bach and Güldner, and the excellent Italian work of Pomini.*

The length of the crank pin is generally determined from the formula

$$p \cdot v = K,$$

in which p is the pressure obtained as follows by multiplying the area of the piston by the mean indicated pressure (for Diesel engines about 7 atmospheres—100 lbs. per square inch) and dividing this product by the projected area of the pin (*i.e.*, diameter multiplied by the length):—

$$p = \frac{A\, p_m}{d\, l},$$

v is the peripheral velocity of the pin in metres per second.

K is a constant depending on the type of pin, the system of lubrication, the workmanship, etc. For four-cycle engines of normal speed, the following values have been collected from several engines built:—

K = 14 to 16 for crank-shaft journals with ring lubrication.

K = 25 to 30 for crank pins with centrifugal lubrication.

For high-speed engines with forced lubrication,

K = 30 to 40 for the main bearings,

and K = 50 to 60 for the crank pins.

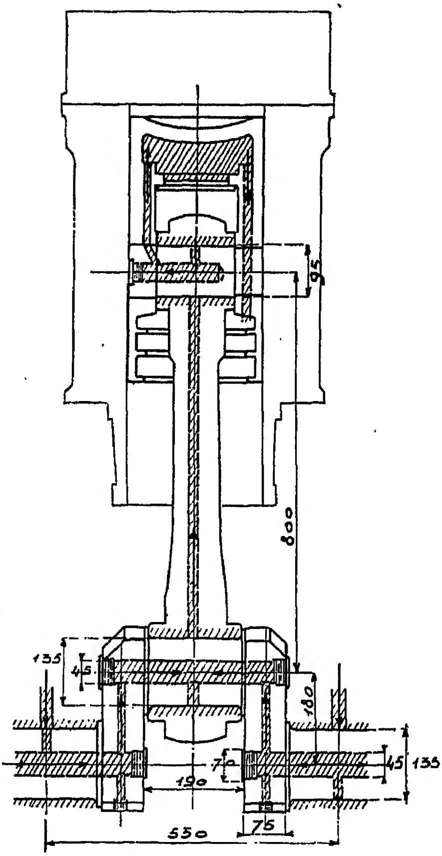


Fig. 115.—Arrangement of Forced Lubrication of Crank Shaft and Gudgeon Pin, and Oil Cooling of Piston Crown (M.A.N.).

For two-cycle engines, slightly lower values are generally assigned to K.

The constant for the crank pin is always greater than that for the journals, although the lubrication of the latter is easier and generally more efficient.

* O. Pomini, *Piston Engines.* Milan, Hoepli, 1912.

The explanation may be found in the efficient air cooling which rotation gives to the crank-pin bearing.

The distance between the centres of the main bearings in ordinary engines is approximately two and a-half times the diameter of the cylinder with ring lubrication, whilst with forced lubrication the distance between bearing centres may be less than two cylinder diameters.

In marine engines the thrust block bearing area is calculated on the basis of a mean pressure on the thrust shoes, of 3 to 4 kgs. per square cm. (40 to

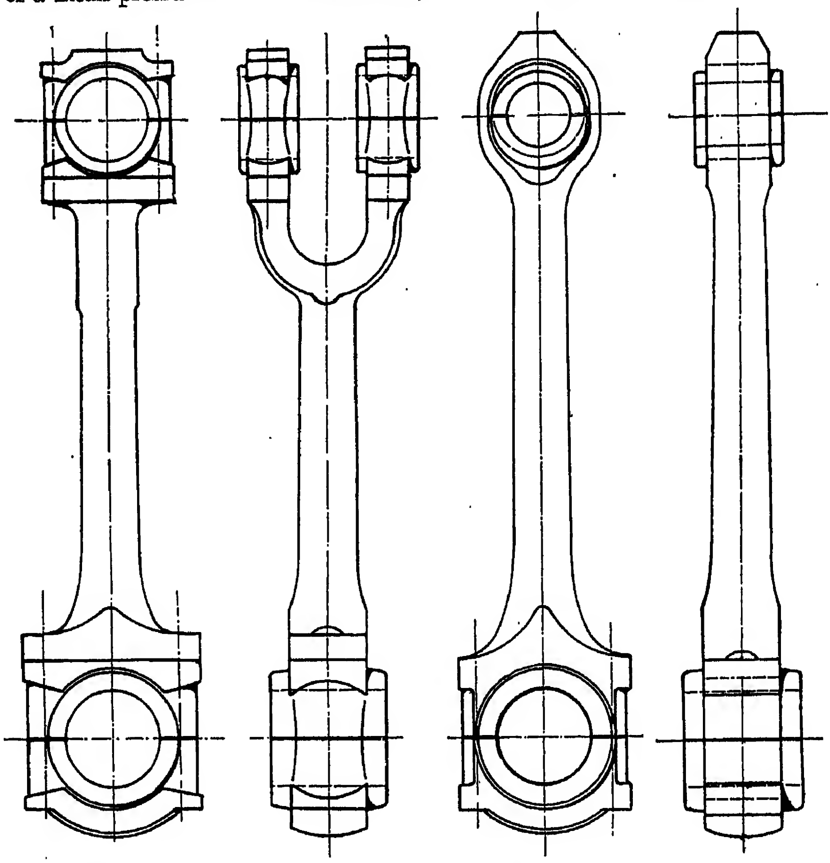


Fig. 116. Fig. 117. Fig. 118. Fig. 119.

Figs. 116 to 119.—Types of Connecting-rods.

60 lbs. per square inch) in the case of cargo boats, up to 6 or 7 kgs. per square cm. (85 to 100 lbs. per square inch) for torpedo boats and submersibles,* keeping the diameter of the collars about 1·6 that of the shaft, and the thickness 0·14 to 0·20 of the same diameter.

* With thrust blocks of the Maudslay type, only the upper half of the shoe is taken into account for the calculation of the thrust area.

In calculations to a near approximation it may be considered that the thrust area should be about 8 square cm. (1·25 square inches) for every 10 B.H.P. of the engine.

Connecting-rods.—In Diesel engines these are always of forged steel, tapered and sometimes hollow, especially with light engines. The bottom end is of the marine type, generally fitted with cast iron, cast steel, or gun-metal bushes lined with white metal. The crank-pin bearing bolts are made fairly strong, although not subjected to any considerable stress. They are not in tension, except when arresting the reciprocating masses at the end of the exhaust stroke of four-cycle engines.

Figs. 116 and 117 show the type of connecting-rod for a crosshead engine, and Figs. 118 and 119 that used with an engine having the gudgeon pin inside the piston.

The top and bottom ends may be forged with the body of the rod (Figs. 118 and 119), or may be independent (Figs. 116 and 117).

In the first case, the turned bushes of bronze, cast iron or cast steel, lined with white metal, are separate from the jaws of the rod, and with the second the white metal is generally run directly into the two half-brasses forming the head.

Complete top and bottom end bearings are generally provided as spare gear, to serve in the event of one on the engine having to be removed for the renewal of its white metal due to overheating.

The type shown in Figs. 116 and 117 is not quite so rigid perhaps, but offers the advantage that it is possible to alter at will the distance between the crank pin and the gudgeon pin, by inserting liners of sheet metal between the foot of the rod and the brass. With Diesel engines it is very useful to be able to alter the length of the connecting-rod in this way, as it is the most convenient method of varying the final cylinder compression pressure.

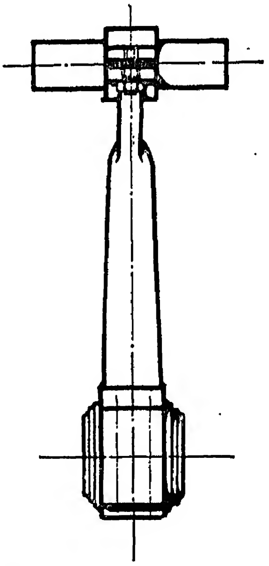


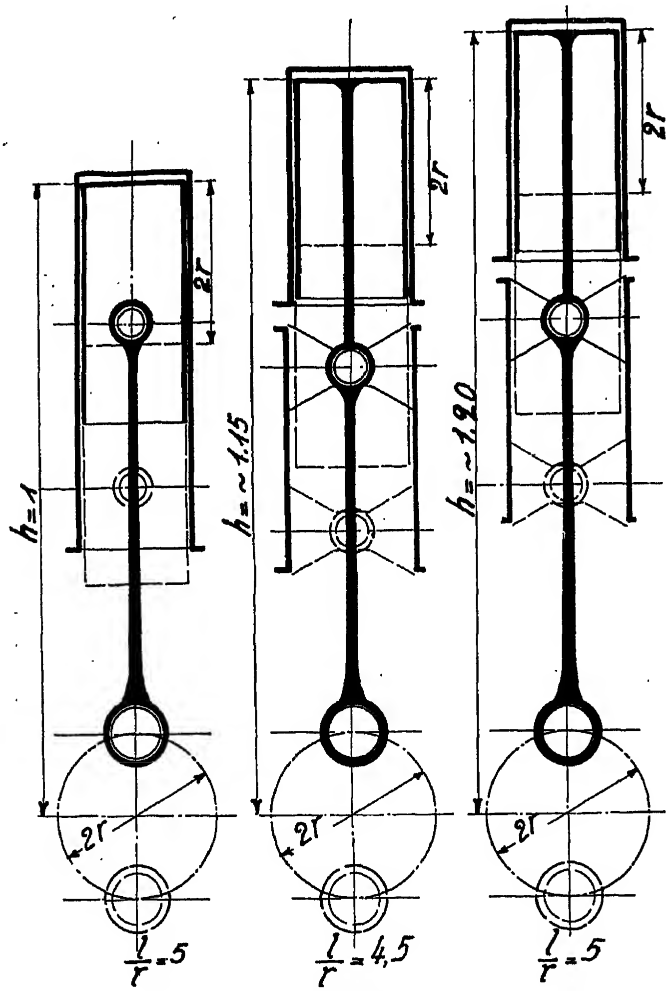
Fig. 120.—Connecting-rod, with Gudgeon Pin rigidly attached (Sabathé).

Fig. 120 shows a connecting-rod to which the gudgeon pin is rigidly attached, the latter being free to move in the bosses of the piston, which in this case are fitted with the bearing brasses. Imperfect alignment between the axis of the cylinder and that of the crank has a greater influence in this case upon the running of the bearing, and for this reason the top end of the connecting-rod is flattened out to reduce its rigidity in the longitudinal direction of the shaft (Sabathé).

The ratio $\frac{l}{r}$ between the length of the connecting-rod and the radius of the crank is generally about 5, but may be reduced to even 4 for high-speed or marine engines where the maximum reduction of height is desirable. When a crosshead is fitted a ratio $\frac{l}{r} = 4{\cdot}5$ may be considered sufficient even

for ordinary engines, because the horizontal component N (Fig. 140 on p. 108) acts on the guides, where the wear is less harmful, instead of on the cylinder walls.

The crosshead does not call for particular mention, except that the dimensions of the pins should be large, in order to reduce the pressures on the brasses.



Figs. 121 to 123.—Comparison of Heights of Engines with and without Crosshead.

One of the main reasons for adopting a crosshead is to obtain a reduction of these pressures, since, with the top end of the connecting-rod enclosed in the trunk piston, its dimensions are limited by those of the inside of the piston. Further, it may be added that, with a crosshead, lubrication is more easily carried out; the parts are cooled by their movement, and are further

removed from the radiation of the hot walls of the piston. In modern practice a crosshead is almost without exception adopted with large engines, for the above and other very good reasons.

The trunk piston has two important functions, that of keeping the cylinder gas tight, and that of serving as a guide. When the piston is large and the stresses considerable, each of these duties assumes such importance that it is difficult for one part of the engine satisfactorily to fulfil both. Güldner states that with large pistons the difficulties in fulfilling satisfactorily the principal function—*i.e.*, that of an obturator—are such that it is unwise to make it a heavily loaded guide as well. Large pistons require to take the thrust on a relatively smaller area than do small ones; otherwise the piston dimensions would assume such values as would cause excessive weight, expansions and stresses in the material. Difficulty would also be experienced in lubricating the large rubbing surfaces, which require perfect and abundant lubrication on account of the large loads upon them.

The disadvantages attendant upon the adoption of the crosshead may be summed up as an increase of weight, of cost, and especially of height

Fig. 124.

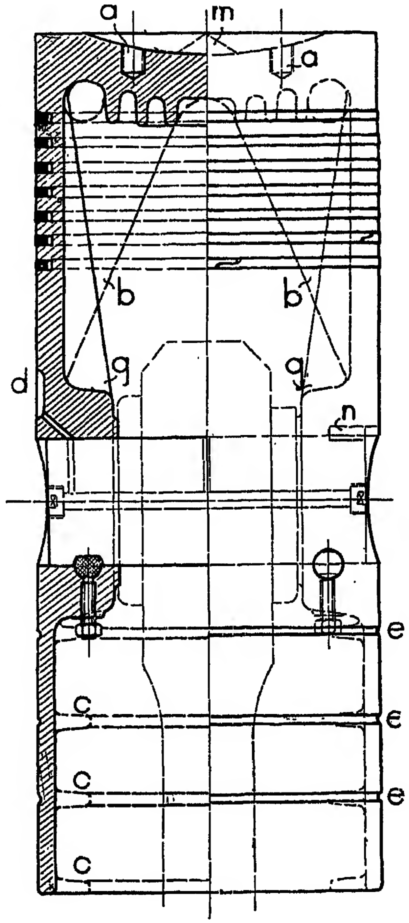


Fig. 125.

Figs. 124 and 125.—Simplest Type of Piston.

of the engine. For an equal ratio $\frac{l}{r}$ the connecting-rod, piston-rod and crosshead give an increase of about 20 per cent. in height (Figs. 121 and 123), but, as stated, the better conditions under which the crosshead slippers work, make it possible, without appreciable disadvantage, to shorten slightly

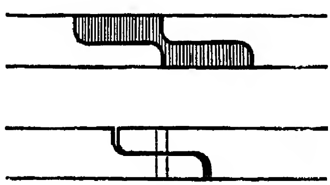
the connecting-rod, so that the increase of height may be usually considered as somewhat below 20 per cent.

Pistons.—The simplest type of piston for Diesel engines is shown in Figs. 124 and 125.

Being cast in one piece and without special cooling arrangements for the crown, it is suitable for small and medium-sized units, particularly those working on the four-stroke cycle. With two-cycle engines, even of small diameter, it is preferable artificially to cool the crown of the piston. The part of the piston wall from the crown to the gudgeon-pin bosses is thick, whilst the lower part towards the open end is thinner and often strengthened by ribs *c*.

The rings, to the number of six or seven, are of the best cast iron, generally square in section. They should be turned twice; firstly, before being cut, they are turned to their free diameter; then cut as shown in Fig. 126, the sectioned part removed, and the ring closed by a small pin, as shown in Fig. 127; secondly, they are turned to the diameter of the cylinder. To keep the joints of the various rings out of line, a small radial pin registering with a hole in the ring is screwed into the bottom of each groove.

The gudgeon pin, of case-hardened steel finished by grinding, is fixed in the piston by two set pins. Sometimes the ends of the pin in the bosses of the piston are slightly conical, and in this case the small end of the cone is screwed and fitted with a thin nut bearing against the piston and retaining the pin in place. The small key *n* prevents the gudgeon pin from turning. The bosses in the piston are usually strengthened with ribs *g*, as shown in Fig. 125.



Figs. 126 and 127.—Piston Ring Joint.

The external diameter of the main part of the piston is only some one-tenth of a mm. ($\frac{4}{1000}$ of an inch) less than that of the cylinder; the upper part carrying the rings is usually from two-tenths to five-tenths of a mm. ($\frac{8}{1000}$ to $\frac{20}{1000}$ of an inch) smaller than the lower part.*

The piston crown is sometimes flat, but often concave in form, and is always very thick. It is generally strengthened by ribs (Fig. 125), either of annular form as shown in section, or radial as shown dotted at *b*. If the concavity of the crown is great, in order to keep the compression volume at the correct value, the piston crown will require to come very close to the head of the cylinder at the end of the stroke, in which case recesses at the periphery of the piston crown are left to provide room for the valves (Fig. 128).

The tapped holes *a* (Fig. 125) are for eye bolts, to which the lifting tackle for withdrawing the piston is attached.

The conditions under which Diesel engine piston crowns work are very severe, as can be readily understood by considering the simultaneous action of the very high temperatures and pressures to which they are subjected. On

* These values are for small engines and increase with the size of the engine though not in proportion thereto.

this account, in preparing designs, trials with various forms of strengthening and thicknesses, depending upon the quality of cast iron, are more important than accurate calculations. The thicknesses should not be greater than necessary, in order that the weight of the reciprocating masses may be a minimum, and that over-heating of the crown, and the formation of stresses, due to large differences of temperature in the material of the crown itself, may be prevented. It is better to have recourse to the use of the very best material, careful casting, and rational strengthening by ribs.

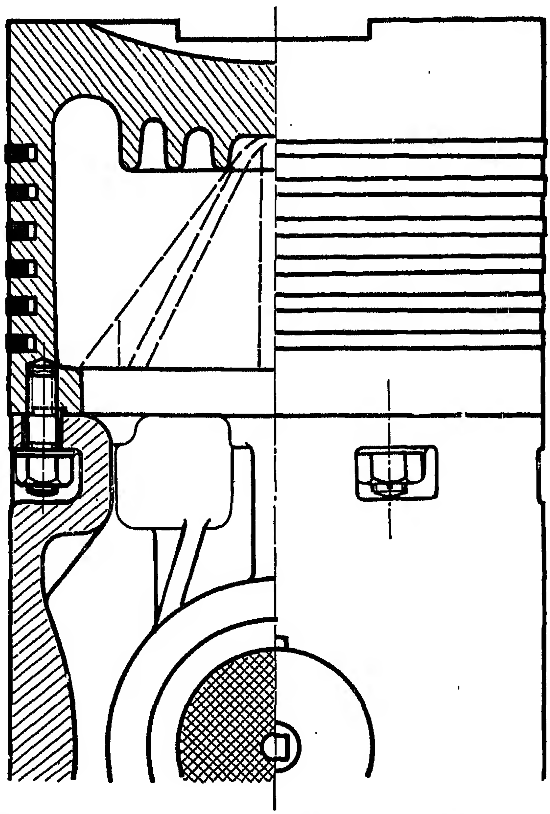


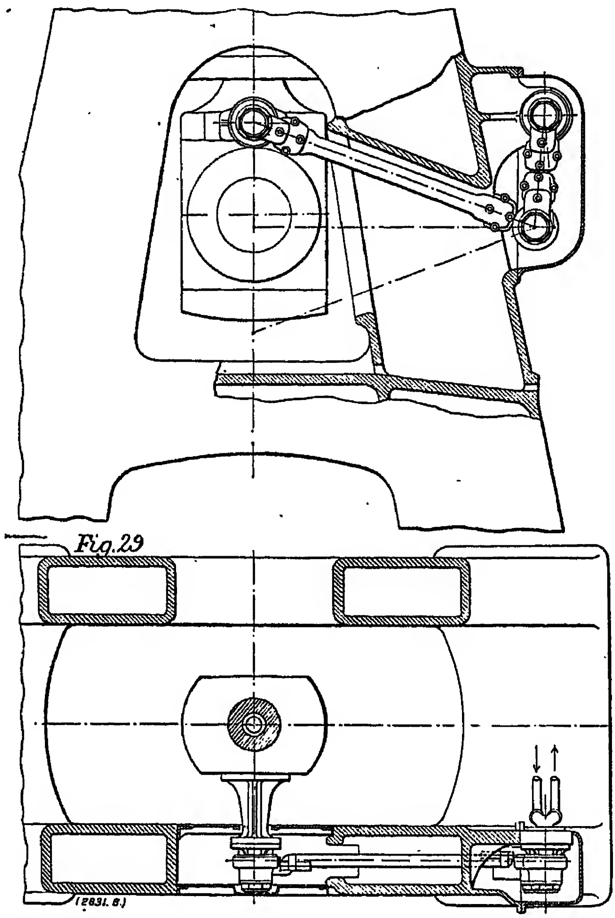
Fig. 128.—Piston with separate top portion (M.A.N.) having recesses in Periphery of Crown to give Clearance for Valves in Cylinder Head.

Above a certain limit of diameter (generally 300 to 350 mm.—12 to 14 inches—the piston is frequently made in two pieces (Fig. 128), giving rise to a certain increase in weight and cost, but facilitating casting, reducing the internal stresses due to expansion, and permitting of greater freedom in assigning the most suitable thicknesses to the various parts.

Amongst the expedients adopted to improve the condition of the piston crown may be cited one of placing in the centre of the crown a disc of nickel

steel of a diameter of about one-third that of the cylinder (Sulzer), or of boring out the centre of the crown, and plugging up the hole with a large stud, which tends to reduce the expansion stresses (M.A.N., Figs. 138 and 139, p. 107).

Calculations for this part, although of relatively small moment for the



Figs. 129 and 130.—Joint of Piston Water-cooling System (Tosi).

reasons given, are generally on a basis of a uniformly loaded flat or concave plate supported at its circumference.

Where the ribs cannot be said to afford much support, as is generally the case with water or oil-cooled pistons (Figs. 136 to 139), their strengthening influence is usually neglected.

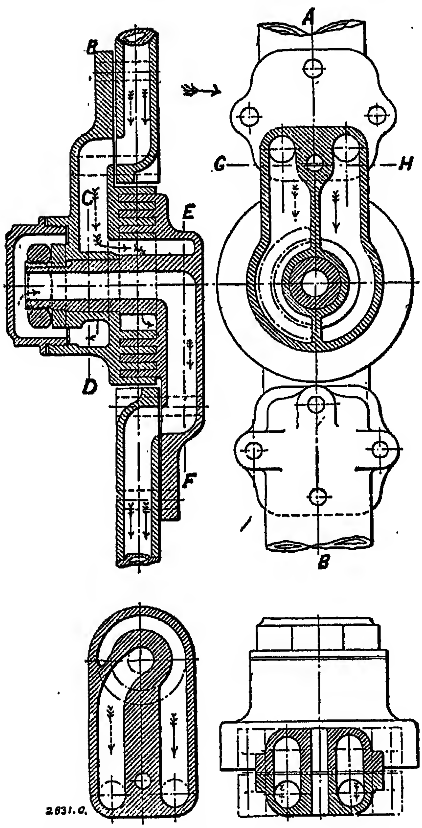
For simple pistons the thickness S of the crown is

$$S = 0{\cdot}12 \text{ to } 0{\cdot}14\,D$$

for large to small diameters (D) of the cylinder.

For high-powered engines, especially those working on the two-stroke cycle, and high-speed marine engines even of small cylinder diameter, the pistons should be artificially cooled. This may be effected by means of air, oil, or water.

Air cooling has the virtue of simplicity, but is not very efficacious. The



Figs. 131 to 134.—Details of Joint of Piston Water-cooling System (Tosi).

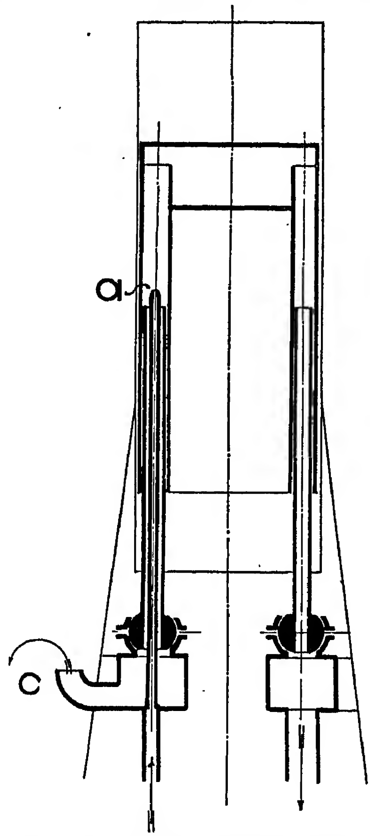


Fig. 135.—Diagram of Sulzer Water-cooling System for Piston Crown.

Werkspoor slow-running four-cycle engines of relatively low power of the M.V. "Vulcanus" do not require much cooling of the piston crowns, and the delivery pipe of a centrifugal fan branches into a number of telescopic tubes, which blow a current of air into the interior of each piston, whence it escapes to the atmosphere.

When water cooling is adopted, the water is led to the piston crowns through hinged or telescopic tubes (Figs. 129 to 134, Tosi). The leaks and

obstructions which may easily occur at the joints constitute the greatest drawback of this system.

Generally the water service is under a pressure of 3 to 4 atmospheres (43 to 56 lbs. per square inch), but to reduce the probability of leaks, Messrs. Sulzer have adopted a system of open pipes represented diagrammatically in Fig. 135. In this system a central tube *a* directs a jet of water against the crown of the piston. The water falls back into the circulating chamber and flows away through a telescopic pipe. The central tube *a* is enclosed within another telescopic pipe, which ends in an overflow *c*.

Figs. 136 and 137 show two pistons with cooled crowns for cargo boat engines; the first is that of the M.V. "Monte Penedo," the second that of the M.V. "Evestone."

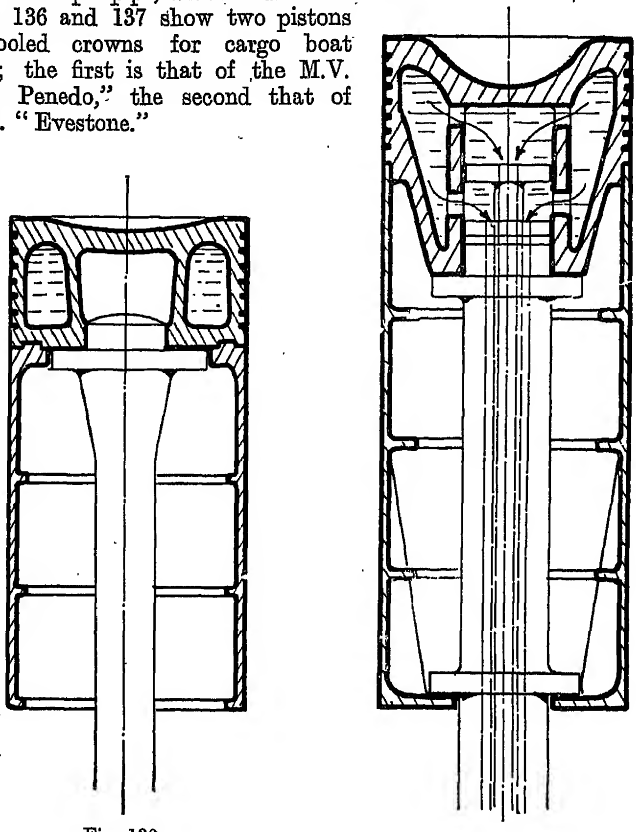


Fig. 136. Fig. 137.

Figs. 136 and 137.—Water-cooled Pistons for Cargo Boat Engines.

Even with marine engines fresh water is sometimes used for piston cooling.

With two-cycle crosshead engines the total length of the piston is considerable, although the piston proper only consists of the upper part with thick walls, in which the rings are carried, whilst the lower thin and light part really serves as a shroud, to prevent the exhaust ports being uncovered

and put in communication with the engine room when the crank is near the top dead centre.

The centre part of the crown is not water-cooled in the piston shown in Figs. 138 and 139 (M.A.N.), which represents a stepped piston, the lower part of which serves as a scavenging pump.

The part of the piston where expansion may give rise to the most dangerous internal stresses is close to the joint between the crown and the cylindrical walls, as the internal fillets lead inevitably to an increase of thickness, tending to store up the heat. In addition, excessive heating of this part is detrimental to the life and working of the uppermost rings, which are always subject to the most severe conditions.

On the other hand, it is not advisable to lower the temperature of the

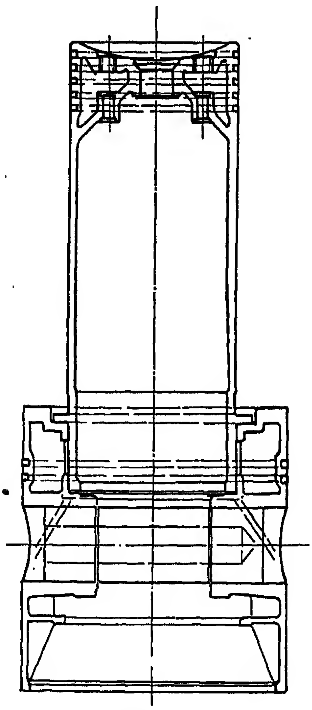


Fig. 138.

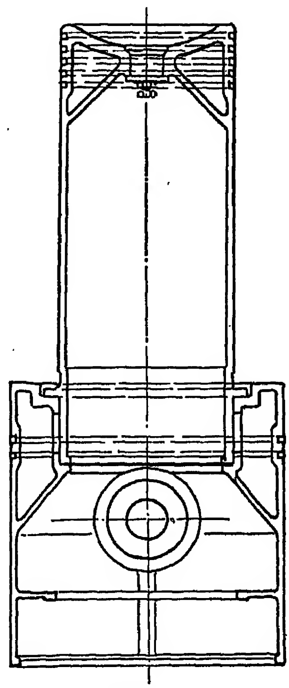


Fig. 139.

Figs. 138 and 139.—Stepped Piston of M.A.N. Two-cycle Engine.

centre of the crown to an excessive extent, because the injected fuel ignites more readily and burns more completely when it strikes against a hot wall.

Reference was made on p. 102 to the fact that the upper part of the piston carrying the rings is always less in diameter by some tenths of a mm. than is the remaining part of the piston. This practice is due to the necessity of compensating for the effects of the varying expansions in the parts of the piston at different temperatures. When the crown is oil- or water-cooled, this reduction of diameter may be less. By adjusting the quantity of cooling

water, the expansions of the various parts of the piston may be regulated to some extent.

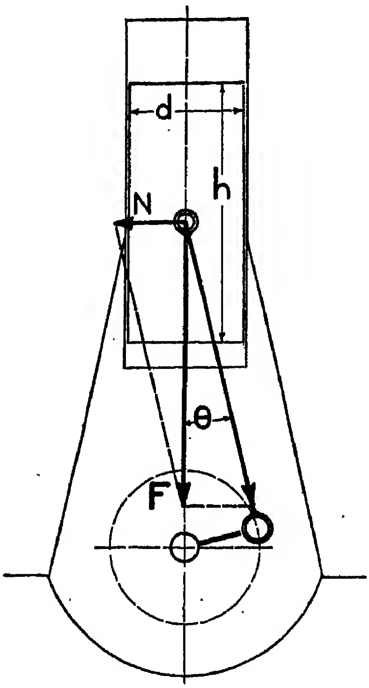


Fig. 140.—Diagram of Forces showing Side Thrust due to Obliquity of Connecting-rod.

Piston-cooling by oil is preferably adopted with high-speed and light engines. Mention has already been made on p. 96, Fig. 115, of the general lines of this application, and the advantages peculiar to this method are—the elimination of the hinged pipes for supply and discharge, and the fact that leakage will not give rise to rusting of bearings or the lathering of the oil in the sump of the bed plate, as would be the case with a leakage of cooling water.

A crosshead absorbs the thrust N (Fig. 140) due to the obliquity of the connecting-rod. When no crosshead is fitted this thrust is taken by the cylinder. For this reason, the total length h (Fig. 142) of the piston should be such that the mean pressure on the area $d \times h$, due to the force N, is not greater than a predetermined value.

Fig. 141 gives the diagram of forces for a ratio of length of connecting-rod to crank radius of 5, and for normal cylinder pressures.

On C_c and C_e as abscissæ, the compression and expansion strokes are developed. In the other two strokes of the cycle the force N may be neglected. The value of the force N

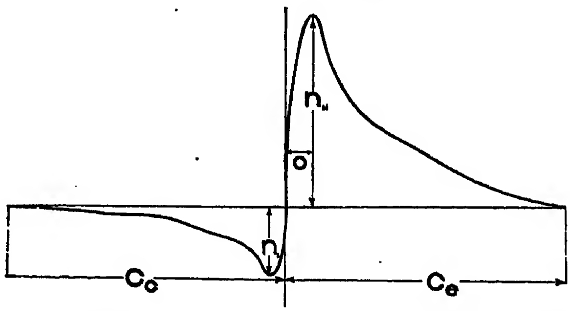


Fig. 141.—Diagram of Value of Side Thrust due to Obliquity of the Connecting-rod at any point of the Stroke.

referred to 1 sq. cm. of the area of the bearing surface of the engine piston are the ordinates. These ordinates, n, are of opposite sign for the

two strokes, since the forces act upon diametrically opposite sides of the piston.

Since $$N = F \tan \theta = A\, p \tan \theta,$$

in which A is the area of the piston of a diameter D, and p the total pressure on the piston, the ordinates of the curve are :—

$$n = \frac{A}{N} = p \tan \theta.$$

During the compression stroke the maximum pressure $n_{,}$ is equal to about 1·2 kgs. per sq. cm. (17 lbs. per square inch), and for the expansion stroke $n_{,,}$ about 3·4 kgs. per sq. cm. (50 lbs. per square inch), and these occur at a distance o (Fig. 141) from the top dead centre—*i.e.*, about one-tenth of the stroke.

The maximum pressure K between the piston and the liner occurs at about one-tenth of the expansion stroke, and has the value

$$K = \frac{n_{,,}\frac{\pi}{4}d^2}{h\,d} = \frac{3{\cdot}4\frac{\pi}{4}d^2}{h\,d} = \text{about } 2{\cdot}7\,\frac{d}{h},$$

from which $$h = \text{about } \frac{2{\cdot}7d}{K}.$$

Usually the ratio $\frac{h}{d}$ (Fig. 142) varies from 2·5 for small to 1·9 for large engines, and the value of K varies correspondingly from 1·1 to 1·4 kgs. per sq. cm. (16 to 20 lbs. per square inch).

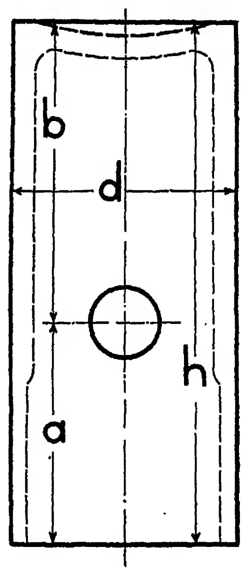


Fig. 142.—Piston Dimensions.

Rational rules cannot be given to determine the position of the gudgeon pin—*i.e.*, of the ratio between the lengths b and a (Fig. 142). A small value of b should reduce the total height of the engine, but leads to the disadvantage that the top end of the connecting-rod is brought nearer to the piston crown, and would be subject to higher temperatures. On the other hand, for a larger value of b, relative to a, the gudgeon pin bearing works under better conditions, but the height of the engine is increased.

Generally, the ratio $\frac{b}{a}$ is from 1·1 to 1·4, and is often 1·30 to 1·35.

Endeavours have been made with trunk piston engines to reduce the wear of the pistons and liners due to the force N, wear which necessitates re-boring the liner and changing the pistons after the engine has been in use for a certain time.

One arrangement for attaining this end consists of fitting white metal slippers to the piston at that part where the force N is greatest. The result of this system is good, though if the white metal be doing its work, and on that account projecting slightly above the surface of the piston, the latter is not so well guided, and may rock slightly when the engine is

running. For this reason, when fitting the white metal slippers, the amount of their projection (a fraction of a mm.) must be adjusted with the greatest care. This system, however, does not permit of the piston length being reduced, since its guiding effect must be retained.

The M.A.N. firm have made use of cast-iron packing pieces let into the piston at that part where N is small, and by adjusting these packing pieces by wedges the play between the packing pieces and the liner may be maintained at any desired value, although the liner may be worn oval. In this case the object is not so much to reduce wear as to compensate for its effects.

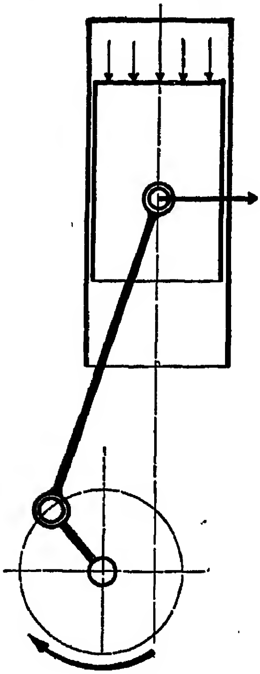


Fig. 143.

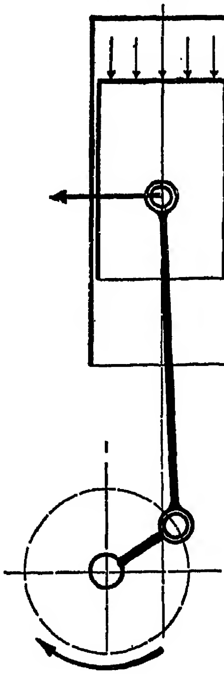


Fig. 144.

Figs. 143 and 144.—Cylinder Offset relative to Crank Shaft to reduce Wear on Liners.

The wear of the pistons and liners may also be reduced by displacing the centre line of the cylinder relative to that of the shaft (as in Figs. 143 and 144), by which means the angle θ, during the expansion stroke when the pressure in the cylinder is greatest, is kept small, and the product $p \tan \theta = N$ is reduced.

This system (having those advantages peculiar to long connecting-rods without the disadvantage of increasing the distance between the cylinder head and the crank shaft) is applied to vertical engines, especially those working with petrol and paraffin, and also to horizontal gas engines (S.L.M. Winterthur), but has only been adopted by one maker of Diesel engines.

The condition of minimum wear is obtained with an off-setting, which gives equal friction between the piston and cylinder walls during the com-

pression and expansion strokes. This is obtained with such a displacement of the centre line of the cylinder as to make equal the areas above and below the axis of the abscissæ in the diagram (Fig. 141), and the liner would then wear oval on the two sides symmetrically, instead of almost entirely on that side on which N acts during the expansion stroke. Consequently, the increase of diameter on re-boring is considerably less.*

With engines of the crosshead type, either one or two slippers (even with reversible engines) are provided. In the former case the slipper is on that side where the pressure $n_{,,}$ (Fig. 141) acts when going ahead. The pressures $n_{,}$ when going ahead and $n_{,,}$ when going astern are taken by the astern guides.

The area Ω of the slipper is calculated on the basis of a mean pressure K of 3 or 4 kgs. per sq. cm. (43 or 56 lbs. per square inch).

$$\Omega = \frac{n_{,,}\frac{\pi}{4}d^2}{K} = \frac{3\cdot4\frac{\pi}{4}d^2}{3 \text{ to } 4} = \text{about } (0\cdot9 \text{ to } 0\cdot7)\ d^2,$$

in which d is the cylinder diameter.

Since with four-cycle crosshead engines the piston serves no other purpose than that of an obturator, the length may be reduced to that necessary to carry the six or seven spring rings required for tightness.

With two-cycle engines the piston, as already stated, has also to keep the exhaust ports covered during the inward stroke of the piston, and for this reason its total length should be about that of the stroke.

When designing Diesel engines, it must not be forgotten that piston cleaning is one of the operations forming part of the routine of engine maintenance, and on that account this piece should be able to be easily disconnected and removed.

One of the advantages of horizontal engines (p. 42) is the readiness with which the piston may be removed from the crank shaft end of the cylinder.

In vertical engines, where the pistons are of a single diameter—*i.e.*, not stepped—they are usually removed from above. The piping connected to the head being removed and the head itself lifted off—the lifting bolts provided for the purpose being screwed into the holes *a* (Fig. 125)—and the bottom end of the connecting-rod being disconnected, the piston and connecting-rod are lifted out by means of suitable gear.†

This system is not altogether the best, since removing the head means disconnecting a great number of pipes, and in the case of marine engines cannot be applied in all cases, because the lifting tackle has to be hooked into an eyebolt situated at a height of almost double that of the engine, which height is not always available in the engine room.

In some small high-speed and marine engines, especially with the cylinder

* It is known from the theory of machines that, with the centre line of the piston displaced relative to that of the shaft, the stroke is greater than $2r$, and the time occupied by the outward differs from that of the inward stroke. For a small displacement of the centres, these factors are negligible.

† In designing the bottom end of the connecting-rod it must not be forgotten that it has to pass through the cylinder: this is not always easy.

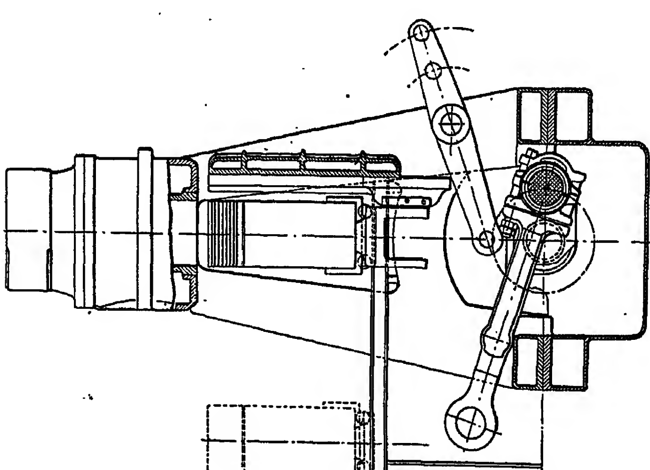


Fig. 145.—Arrangement for removing Piston from below (Tosi).

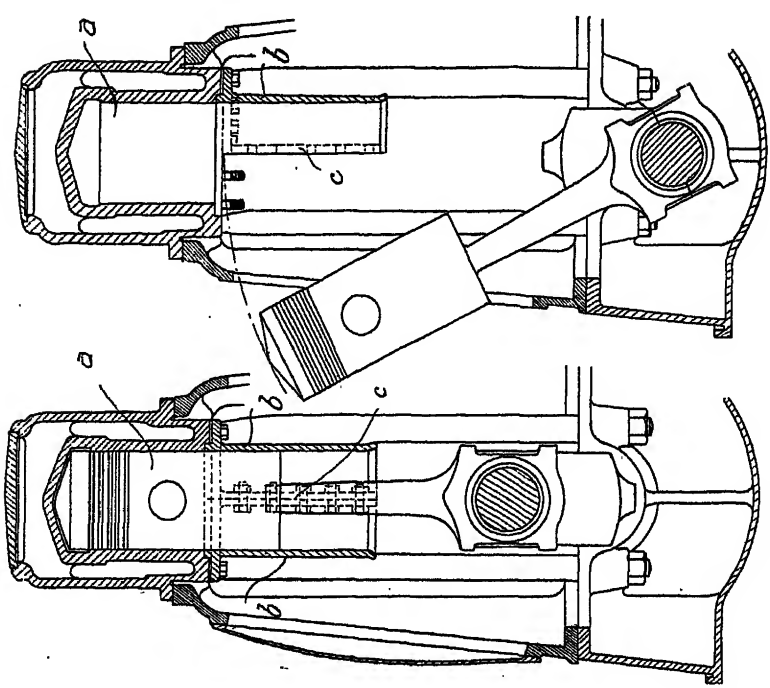


Fig. 146.

Fig. 147.

Figs. 146 and 147.—Method of removing Piston from below (Werkspoor).

and the head cast in one piece, the piston is removed from below,* or the whole cylinder is lifted for examining the piston. This last operation must generally be performed if the piston be of two diameters, of which the larger serves as a scavenging pump piston (Figs. 138 and 139, p. 107).

With crosshead engines, the frame frequently permits of the withdrawing of the piston to the side after it has been removed from the cylinder from below, as shown in Fig. 145 (Tosi).

The Werkspoor firm has given the greatest consideration to the problem of easy removal of the piston. Figs. 146, 147, and 148 clearly explain the two patented methods adopted in the engines of this firm. As shown in Figs. 146 and 147, the lower part *b* of the cylinder is separate from the water

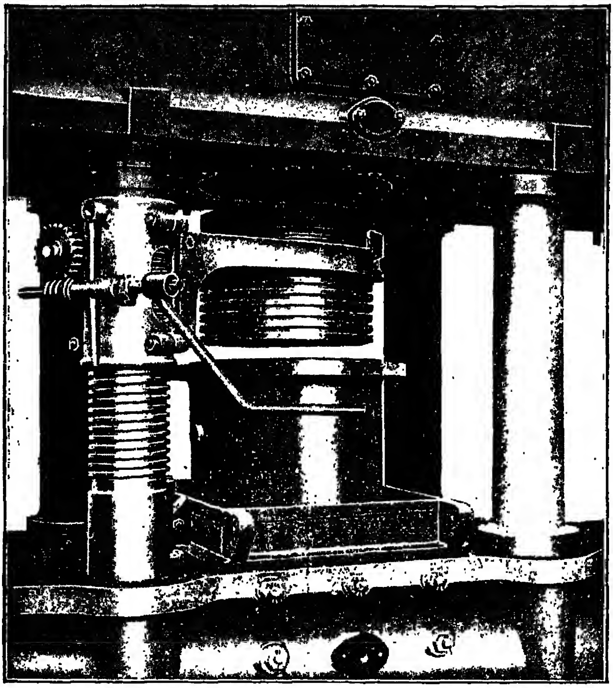


Fig. 148.—Method of removing Piston from below (Werkspoor).

jacketted upper part *a*, and is divided into two halves vertically. By removing the front half it is possible, when the crank is at the bottom dead centre, to swing the piston forward out of the cylinder.

With the system shown in Fig. 148, the lower part of the cylinder is also independent of the upper part, but is in one piece. Two lifting arms, carried

* The bottom end of the connecting-rod is disconnected and the piston is lowered until the gudgeon pin is accessible. This is taken out and the connecting-rod removed, permitting the piston to be completely withdrawn. The operation is not an easy one, and requires some skill.

by a sleeve working on a screw thread cut on one of the turned columns of the engine framing, make it possible to lower, first, the lower part of the cylinder, and, second, the piston.

The Lubrication of Pistons and Gudgeon Pins.—When forced lubrication is adopted for the main bearings, the gudgeon pin may receive its lubricating oil from the same circuit at a pressure of about 2 or 3 atmospheres (30 or 40 lbs. per square inch). For greater safety, many constructors prefer, how-

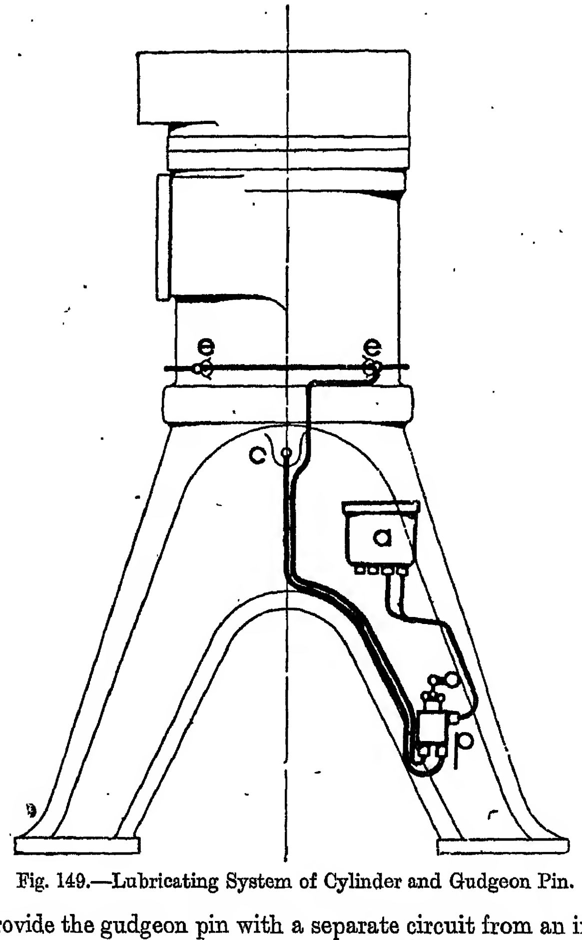


Fig. 149.—Lubricating System of Cylinder and Gudgeon Pin.

ever, to provide the gudgeon pin with a separate circuit from an independent pump through hinged tubes similar to those used for the piston crown cooling water (Figs. 129 and 130 on p. 104).

Part of the lubricating oil for the gudgeon pin is sometimes led through holes to the exterior surface of the piston, and so lubricates the cylinder.

It is well to make provision that the oil escaping from the other moving parts may not mix with that falling from the piston; for, whilst the first after having been cooled may be used again, the second is always partially

burnt, and should not be used again except as fuel, unless it be filtered, mixed with fresh oil and used for the less important bearings. To prevent the burnt oil from mixing with that from the bearings, guards are provided, as seen in Plates XVIII. and XIX. (facing p. 286), showing an M.A.N. marine engine. When the crank-shaft bearings are not provided with forced lubrication, the gudgeon pin and cylinder are lubricated by means of special small pumps *p* (Fig. 149) having two small pistons.

The delivery tube supplying oil to the gudgeon pin discharges through the liner at a point *c* in line with the recess *d* in the piston (Fig. 125 on p. 101). The recess passes in front of the orifice of the tube during the delivery stroke of the pump, and a measured quantity of oil is forced into the gudgeon-pin bearing through holes drilled in the piston and the gudgeon pin, and is finally led to the crank pin bearing.

The second of the two delivery tubes, which serves to lubricate the cylinder, is connected to four or six nipples *e* (Fig. 149) disposed around the cylinder itself, as shown in the sketch. The most suitable position for these nipples has been discussed on p. 79.

Flywheels for Stationary Engines.—There is nothing peculiar in the construction of flywheels for Diesel engines; they are generally in halves bolted together at the rim and the boss. The boss joint is sometimes strengthened by means of the two steel rings shrunk on. The rim joint may be made with a large pin secured by two cotters, in addition to or instead of with bolts (Fig. 150).

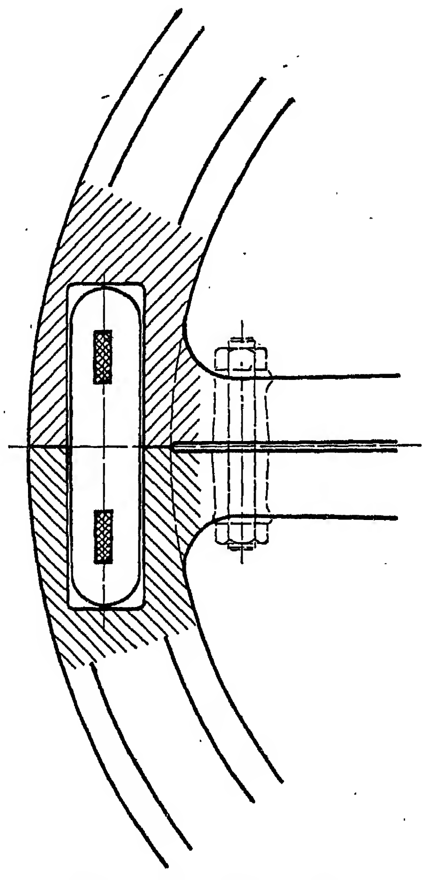


Fig. 150.—Flywheel Rim Joint.

With small engines and a single-piece flywheel, to avoid excessive casting stresses, it is preferable that the boss should be divided by three radial cuts at 120°, closed with the usual shrunk-on steel rings.

The arms are usually six in number, or for large engines eight; when the rim is very large, some constructors adopt a double series of arms or two identical flywheels side by side.

The flywheel is secured to the shaft by two tangential keys at 90°, or by one or two keys of square section.

The rim carries the teeth in which the ratchet gear engages for turning the engine by hand, and when the flywheel serves as a driving pulley, the teeth are formed on the inside of the rim.

Of all engines, the four-cycle Diesel engine is that in which the irregularity of the cycle is the greatest,* or, in other words, requires the greatest momentum in its revolving masses in order not to exceed a given irregularity of running, and hence the necessity for multiplying the number of cylinders, when with one alone a flywheel of excessive weight would be required.

Single-cylinder engines do not usually exceed 70 B.H.P., for from 60 to 200 B.H.P. two-cylinder engines are frequently adopted, and for 150 B.H.P. upward those with three, four, or more cylinders are most suitable.

For the special purpose of reducing the weight of the flywheel very high peripheral speeds are used; generally 30 metres (98·5 feet) per second, and with heavy flywheels, if they do not serve as driving pulleys, speeds up to 34 and 36 metres (110 and 120 feet) per second are adopted.

To find the weight of the rim necessary in order not to exceed a given irregularity of running δ, for a given engine, the usual calculation based on the tangential forces is used, taking into account the influence of the reciprocating masses.

For approximate calculations the following formula is used :—

$$G = \frac{C\,N}{D^2 n^3 \delta},$$

in which

G = the weight of the rim = 0·55 to 0·70 of that of the whole flywheel.

N = B.H.P. of engine.

D = the diameter of the circle drawn through the centres of gravity of the sections of the rim.

δ = the degree of irregularity.

n = revolutions per minute of the engine.

C = is a coefficient deduced from the diagram of tangential forces, and, therefore, influenced by the cylinder pressures, by the reciprocating masses, by the number of cylinders, etc. Constructors determine the value of C for each design of their engines, and then make exact calculations by means of the above formula.

For four-cycle Diesel engines the values of C are from

56 to 62×10^6	for engines with 1 cylinder.		
25 to 27×10^6	,,	2 ,,	(cranks at 360°).
13 to 14×10^6	,,	3 ,,	(cranks at 240°).
$2{\cdot}8$ to $3{\cdot}5 \times 10^6$	,,	4 ,,	(cranks at 180° and 360°).

The higher values apply to the smallest engines.

For four-cycle high-speed engines, the values of C are about from

27 to 30×10^6	for engines with 2 cylinders		(cranks at 360°).
14 to 15×10^6	,,	3 ,,	(cranks at 240°).
about 4×10^6	,,	4 ,,	(cranks at 180° and 360°).

For two-cycle engines, the value of C is very nearly equal to that for four-cycle engines, having twice the number of cylinders.

* *Vide* Author's article in the *Review Il Politecnico*, No. 24, 1910.

The degrees of irregularity usually designed for Diesel plants are

δ = from $\frac{1}{35}$ to $\frac{1}{40}$ with engines for ordinary work.

δ = from $\frac{1}{50}$ to $\frac{1}{60}$ with engines for delicate work (cotton mills, etc.).

δ = from $\frac{1}{70}$ to $\frac{1}{80}$ for belt-driven dynamos.

δ = from $\frac{1}{80}$ to $\frac{1}{100}$ for dynamos coupled directly to the engine, or for belt-driven alternators.

δ = about $\frac{1}{150}$ for direct-coupled alternators.*

Flywheels for Marine Engines.—With the exception of some two-cycle engines of six or more cylinders, marine engines are always provided with a flywheel. The construction and calculations required for its design in no way differ from those given for land engines, excepting that, in order to avoid large diameters unsuitable for the limited space in the engine room, the peripheral speeds are never high, being limited to about 15 metres (50 feet) per second.

Further, the engines being always of the multi-cylinder type, and an irregularity of about $\frac{1}{50}$ being sufficient for screw propulsion, the weight of the flywheel is never very great in proportion to that of the engine.

If sometimes on checking the calculation of the flywheel of an existing engine, it is found that $\delta < \frac{1}{50}$, it should not be attributed to a desire on the part of the designer to increase the regularity of running, but to a wish to provide the engine with a flywheel such as will facilitate starting.

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* It will be observed that the values given for δ are larger (*i.e.*, the irregularity is greater) than those commonly used for steam engines, but it must be remembered that the great regularity of running of the steam plant is not adopted because it is necessary to the running of the driven machine, but because the flywheel is designed with a weight which will give it sufficient momentum to absorb the sudden variations of load. With Diesel engines the indicator diagram is such that in order not to exceed a degree of irregularity that may be fairly high, a flywheel sufficiently heavy for this other purpose is required. (*Vide Il Politecnico*, No. 24, 1910.)

CHAPTER IX.

CYLINDER HEADS, VALVES, AND FUEL INJECTION VALVES.

THE head of the cylinder is a part of the engine of great importance, and, since it contains all the valves and their passages,* is in addition a difficult part to cast. For this reason, except in engines of small power, it is always cast as a separate piece, apart from the cylinder, and secured to the latter by strong studs or bolts.

The material usually employed is cast iron, though examples of heads made of cast steel may be found. The use of this material increases the

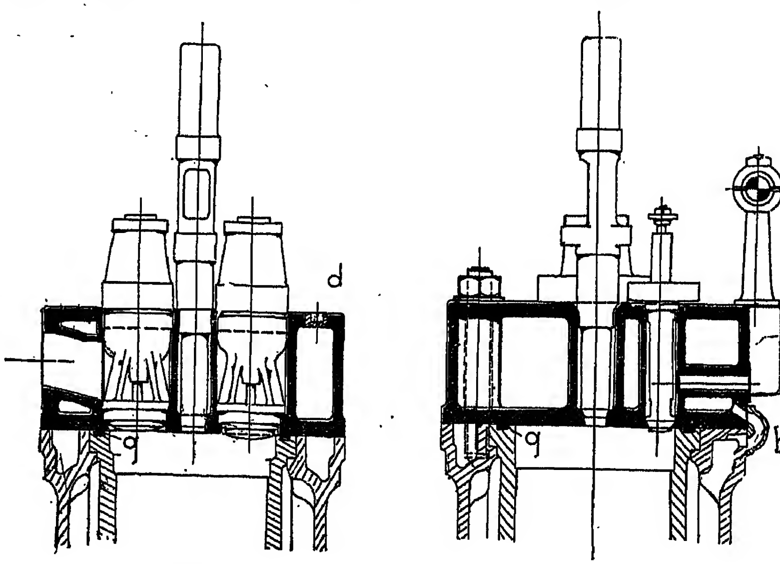


Fig. 151. Fig. 152.

Figs. 151 and 152.—Sections of Cylinder Head of Four-cycle Engine.

difficulties of casting, and its large coefficient of expansion may lead to troubles during the running of the engine.

Cylinder Heads of Stationary Engines.—In four-cycle engines, the head is of cylindrical form, with its top and bottom flat or slightly curved. The spigot for the joint between cylinder and head, shown at "*g*" (Fig. 151)

* Sometimes the starting air valve is at the side of the cylinder just below the head. In the Sulzer engines which have double port scavenging, with the upper ports controlled by a valve, the latter is at the side of the cylinder, as shown in Figs. 37 and 38 on p. 37.

ensures that the latter shall be central with respect to the former; the pin *a* (Fig. 154) makes it certain that the head shall always be replaced with its fore and aft centre line in the same plane as that of the cylinder. The turned pillars C hold the fixed shaft which serves as a fulcrum for the valve levers.

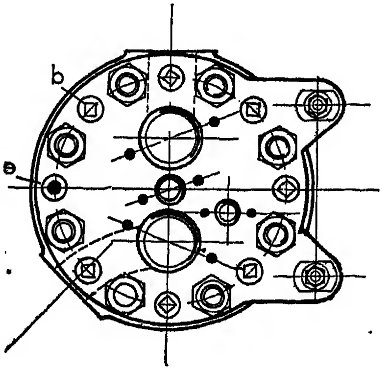


Fig. 153.—Plan of Cylinder Head of Four-cycle Engine.

The suction and exhaust valves and the fuel injection valves are almost always placed on the same diameter; the passages leading to them, however, do not generally enter the head at positions diametrically opposite to one another, for in multi-cylinder engines the space between the adjacent cylinders is not sufficient for the connection of the pipes. For this reason, the flange for the attachment of the exhaust pipe is towards the back of the head (Fig. 153).

In high-speed engines with forced lubrication, in which the space between the cylinders is even more restricted, the suction passage leads from the back of the head (Fig. 156). In these engines, owing to the small cylinder diameter, it also becomes difficult to allow for valves of the necessary dimensions if the suction and exhaust valves are arranged on the same centre line

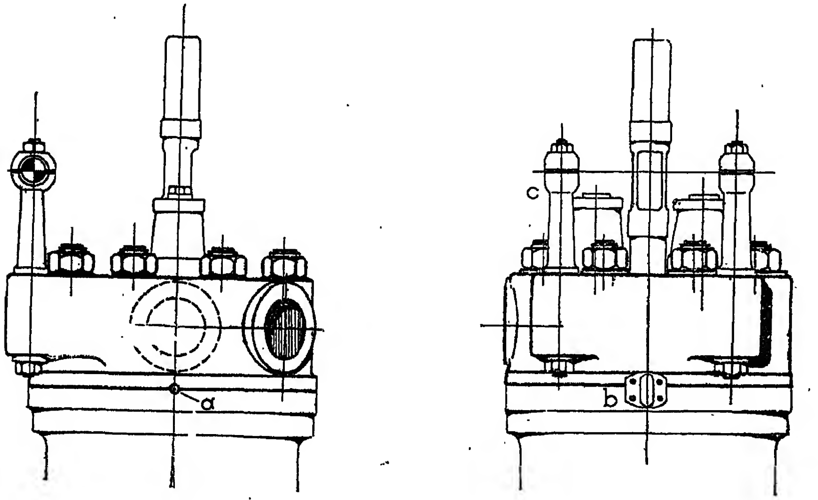


Fig. 154. Fig. 155.

Figs. 154 and 155.—Elevations of a Typical Cylinder Head for a Four-cycle Stationary Engine.

as the fuel injection valve. For this reason it is sometimes necessary to off-set the latter (Fig. 156, Grazer, Sabathé, M.A.N., etc.).

The starting air valve is placed on the middle line at right angles to that of the other valves, or more frequently slightly to one or other side of the middle line, owing to the disposition of the valve-actuating levers.

From the above it is seen that cylinder heads are rarely symmetrical in form, frequently making it necessary to use right- and left-handed patterns.

To be symmetrical, the suction and exhaust passages must be as shown in Fig. 156, and the starting air valve on the diameter at right angles to the other valves.

The cooling water enters the head at the bottom by means of a connecting-pipe *b* (Figs. 152 and 155), and passes out at the top at a point *e* (Fig. 153) diametrically opposite.

The core holes, generally closed with removable bronze plugs, may serve to clean any deposit or incrustation left by the cooling water in the water spaces. Removable doors of large dimensions are often provided to facilitate this cleaning.

The top and bottom walls of the head, where sufficient support is not given by the metal provided for the valve chambers, are usually connected by vertical radial webs, in which holes are cored to permit of free water circulation.

For simplicity of casting, or due to restrictions of space, the valve chambers

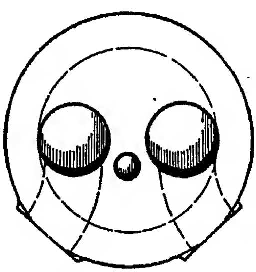


Fig. 156.—Plan of Cylinder Head of High-speed Engine.

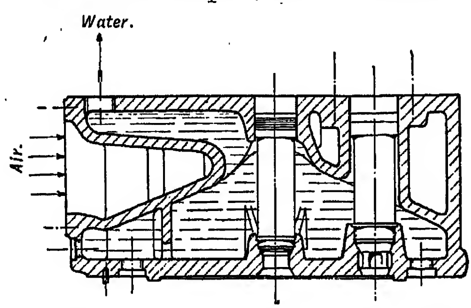


Fig. 157.—Section through Cylinder Head having Valve Chambers separate and secured in the Casting (Tosi).

of the fuel injection valve and the starting air valve, instead of being cast in the head, are sometimes formed by a steel sleeve screwed into the bottom wall of the head, and jointed at the top (Fig. 157).

In cases where space is restricted, the cylinder head studs are encased in copper tubes expanded into the upper and lower walls of the head, instead of passing through cored-out bosses.

The external diameter of the head depends on the design, and is usually greater than 2 D, with a height of 0·7, or more often 0·8 D, where D is the cylinder bore. An even number of studs, eight for medium powers, ten or twelve for high powers, secure the head to the cylinder.

The heads of two-cycle engines differ from those of the four-cycle type in the design of internal passages and spaces, rather than in the external form.

The fuel injection and starting air valves are identical in position and type with those for four-cycle engines, and around the fuel injection valve the two, three, or four scavenging air valves are grouped.

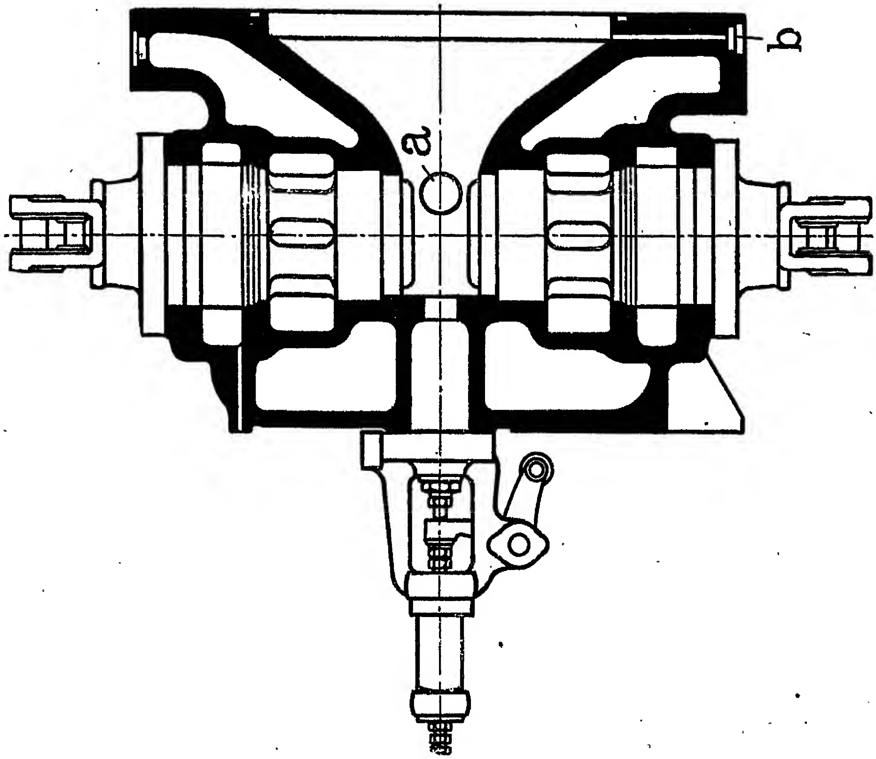
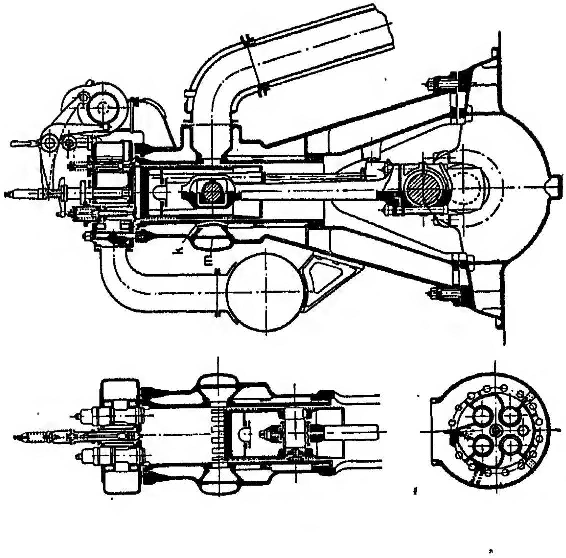


Fig. 161.—Section of Cylinder Head of Two-cycle Horizontal M.A.N. Engine.



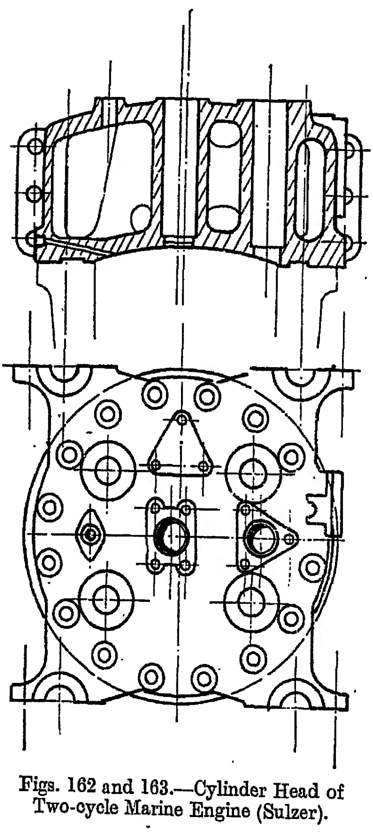
Figs. 158 to 160.—Sections of Two-cycle Sulzer Engine.

The scavenging air is led to the head by a large pipe (Fig. 157, Tosi), and thence to the valve chamber, through a passage of ample cross-section cast in the head, and disposed so that the bottom wall of the head is water-cooled all over its surface.

In Figs. 158 to 160 the plan and section of the head of a two-cycle Sulzer engine are given.

Horizontal engines have heads similar to those of gas engines.

Fig. 161 represents in section a head for a two-cycle horizontal M.A.N. engine. The fuel injection valve is horizontal and placed on the cylinder axis, the scavenging valves are vertical, opposite and superimposed.



Figs. 162 and 163.—Cylinder Head of Two-cycle Marine Engine (Sulzer).

At *a* is seen the starting air valve placed horizontally at the side of the head, and at *b* a drain cock is fitted.

Horizontal four-cycle engines have a head similar to those described for two-cycle engines, except that the lower scavenging valve is replaced by the exhaust valve. The exhaust valve has no cage, since it can be removed through the inlet valve chamber (see Plate XIII., facing p. 240).

Cylinder Heads of Marine Engines.—The description given for the cylinder heads of four-cycle land engines holds good for marine engines working on the same cycle. On the other hand, in the case of two-cycle marine engines, this part assumes many and various forms, according to the number, type and angle of the axes of the valves contained therein.

Some heads only have a fuel injection valve (Figs. 34 to 36, p. 36, Polar), whilst others also have a starting air valve (Figs. 162 and 163, Sulzer). When valve scavenging is adopted, one (Figs. 164 and 165. M.A.N.), two (Figs. 100 and 101, p. 89, Sulzer), or four (Plate XVII., facing p. 272) scavenging valves may be fitted.

From this, necessarily, the internal passages assume different shapes, and the upper and lower walls vary according as the valves are vertical or inclined.

Calculation of Cylinder Heads.—Except in the cases of easily determined stresses, such as those occurring in the cylinder head studs, calculation aids the designer but little, other than to give figures proportioned from experience. Calculations serve to give a conception, more relative than otherwise, which, taken in conjunction with experience of other engines,

and remembering always the exigencies of the foundry, is sufficient for practical purposes.

The head is designed largely as a result of individual opinion, after which

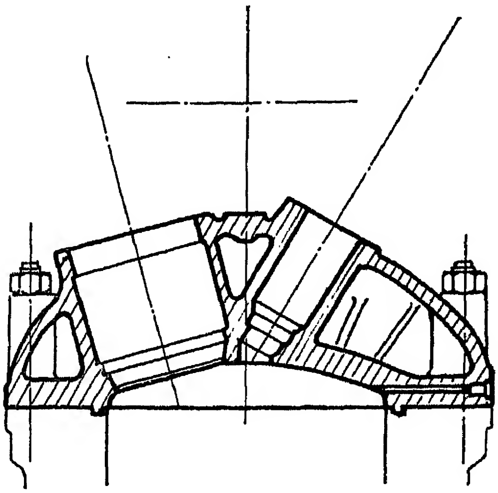


Fig. 164.

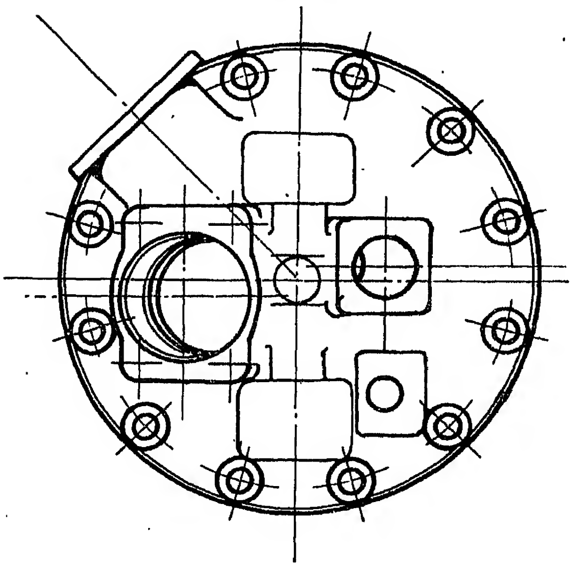


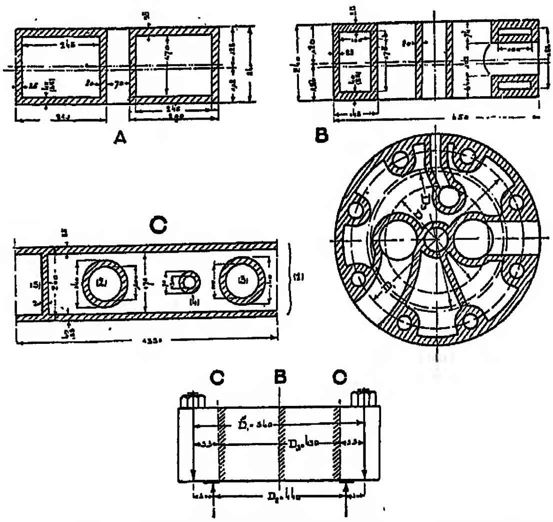
Fig. 165.

Figs. 164 and 165.—Cylinder Head of Two-cycle Marine Engine (M.A.N.).

the thicknesses are checked by calculating the bending moments, whence the stresses are obtained from the moduli of the sections in elevation and in plan.

Of the two walls, the external one, theoretically, should be of the greatest thickness, since the cast iron of which it is made is in tension, due to the bending moments. It is essential, however, to remember that the bottom wall, during the running of the engine, is subject to high temperature. Besides this, the parts of the lower wall enclosed by the external circumferential wall of the head, the radial webs and the walls of the valve pockets, are subject not only to the stresses common to the head as a whole, but also to those consequent upon considering these parts themselves as plates semi-incastrated and subject to the cylinder pressure.

Calculation of the stresses in these areas is impracticable, owing to their irregular form, but allowance is made for them by making the lower wall of a somewhat greater thickness than the upper wall. It is, moreover, impossible to take into account the stresses consequent upon the expansions and contractions due to the differences of temperature between the various parts.



Figs. 166 to 170.—Four-cycle Engine Head dimensioned to illustrate Stress Calculations.

The following calculation to check the dimensions of the head of a four-cycle engine shows the manner in which the stresses which the material has to withstand may be estimated.

Figs. 166 to 170 represent the head dimensioned.

D = the diameter of the cylinder = 330 mm.

D_1 = diameter of the pitch circle of the studs = 540 mm.

D_2 = diameter between the centres of support, in this case the spigot,—assuming the pressure between the joints varies as the distance from the inner surface of the cylinder = 440 mm.

D_3 = the diameter to the inner walls of the bosses surrounding the studs = 430 mm.

To calculate the stresses, taking the maximum pressure at starting, which is assumed as 45 kgs. per sq. cm. (640 lbs. per square inch), the total load on the head is

$$P = \frac{\pi}{4} D_2^2 \cdot p = \text{about } 68{,}500 \text{ kgs.}$$

$$M_f = \frac{1}{2} P \left(\frac{D_1}{\pi} - 0{\cdot}212\, D_2 \right) = \text{about } 2{,}700{,}000 \text{ kg. mm.}$$

To obtain the stress in cross-section A (Fig. 166), taking the top and bottom walls, both 35 mm. thick, with the neutral axis central and 120 mm. from the top and bottom of the head, an error of about 5 per cent. is introduced, but, in view of the rough approximations used throughout this calculation, this is negligible.

$$I = 2 \times \tfrac{1}{12}(290 \times 240^3 - 245 \times 170^3) = 467{,}546{,}000 \text{ mm.}^4$$

$$\frac{I}{Y} = \frac{467{,}546{,}000}{120} = \text{about } 3{,}900{,}000 \text{ mm.}^3$$

The stress in the section A

$$= \frac{M_f}{\frac{I}{Y}} = \frac{2{,}700{,}000}{3{,}900{,}000} = 0{\cdot}7 \text{ kg. mm.}^2$$

To obtain the stress in cross-section B (Fig. 167).

$$I = \tfrac{1}{12}(145 \times 240^3 - 100 \times 188^3 + 100 \times 142^3 - 145 \times 102^3)$$
$$+ 2 \times \tfrac{1}{12}(20 \times 240^3) + \tfrac{1}{12}(145 \times 240^3 - 100 \times 170^3)$$
$$= \text{about } 310{,}000{,}000 \text{ mm.}^4$$

$$\frac{I}{Y} = \frac{310{,}000{,}000}{120} = 2{,}580{,}000 \text{ mm.}^3$$

$$\text{Unit stress on section B} = \frac{2{,}700{,}000}{2{,}580{,}000} = \text{about } 1{\cdot}05 \text{ kgs. mm.}^2 \text{ (1,500 lbs. per sq. in.).}$$

To obtain the stress in cross-section C (Fig. 168); with a structure of such irregularity as a cylinder head, it is not sufficient merely to estimate the stresses in the cross-sections on the diameters; the stresses should also be calculated for a cylindrical section, taking for instance that of diameter D_3, of which the circumference touches the stud bosses and which is most stressed. The section is developed and the load is assumed as concentrated on the circumference of the baricentric circle of $\frac{2}{3}$ D_3 diameter.

The bending moment in the section C is

$$M_f = \frac{\pi}{4} D_3^2 \,.\, P \,.\, \frac{(D_3 - \frac{2}{3} D_3)}{2} = \text{about } 65{,}500 \times 72 = \text{about } 4{,}700{,}000 \text{ kgs. mm.}$$

$$I_1 = \frac{1{,}350 \times 240^3}{12} - \frac{1{,}350 \times 170^3}{12} = \text{about } 1{,}007{,}000{,}000 \text{ mm.}^4$$

$$I_2 + I_3 = 2\{0{\cdot}049\,(140^4 - 100^4)\} = \text{about } 27{,}800{,}000 \text{ mm.}^4$$

$$I_4 = 0{\cdot}049\,(70^4 - 30^4) = \text{about } 1{,}137{,}000 \text{ mm.}^4$$

$$I_5 = \frac{20 \times 170^3}{12} = \text{about } 8{,}188{,}000 \text{ mm.}^4$$

$$Y = 120 \text{ mm.}$$

$$\therefore \quad \frac{I}{Y} = \frac{1{,}044{,}125{,}000}{120} = 8{,}700{,}000 \text{ mm.}^3$$

and the stress in section C

$$= \frac{4{,}700{,}000}{8{,}700{,}000} = \text{about } 0{\cdot}54 \text{ kg. mm.}^2 \text{ (770 lbs. per square inch).}$$

To check the sections B *and* C, *in relation to the stresses due to the tightening up of the nuts.*

With regard to this latter, the head is loaded on the circumference of diameter D_1, and supported on the circumference of the spigot diameter D_2.

Section C.—Supposing the load due to the nuts to be $S = \frac{1}{2} P$, then

$$S = 34{,}250 \text{ kgs.}$$
$$M_f = 34{,}250 \times 50 = 1{,}712{,}500 \text{ kgs. mm.}$$
$$\frac{I}{Y} = 8{,}700{,}000 \text{ mm.}^3$$

The stress in C due to the nuts $= \frac{1{,}712{,}500}{8{,}700{,}000} =$ about 0·2 kg. mm.^{2} (284 lbs. per square inch), so that the total stress in C is

$$0{\cdot}54 + 0{\cdot}2 = 0{\cdot}75 \text{ kg. mm.}^2 \text{ (1,070 lbs. per square inch).}$$

Section B.

$$M_f = \tfrac{1}{2} \,.\, 34{,}250 \left(\frac{D_1}{\pi} - \frac{D_2}{\pi}\right) = \text{about } 545{,}000 \text{ kgs. mm.}^2$$
$$\frac{I}{Y} = 2{,}580{,}000 \text{ mm.}^3$$

Stress in B due to the studs is 0·21 kg. mm.^{2} (300 lbs. per square inch), so that the total stress in B is

$$1{\cdot}05 + 0{\cdot}21 = 1{\cdot}26 \text{ kgs. mm.}^2 \text{ approximately (1,800 lbs. per square inch).}$$

To allow for the effect of the pressure on the lower wall of the head it is given a thickness of 40 mm., leaving the upper wall 35 mm. thick.

In this way the strength of the head, as a whole, is certainly increased and the lower wall of the head has a margin which experience shows to be desirable.

As has already been stated, the unsupported parts of the walls have such an irregular form that the stresses to which they are subjected cannot be calculated in a sufficiently approximate manner.

Valves.—In order that the valves may be removed without lifting off the head, they should be contained in a separate casing or cage. The valve seat, which is conical, or very rarely flat, the guide for the valve spindle, and accommodation for the valve spring are formed in this cage.* Tightness is assured by a conical or flat seating between the cage and the head, and the cage is usually bolted to the head by means of a flange.

The thickness of this flange, and that of the cage, should be ample to prevent deformation, which, on tightening up the nuts, might give rise to a leaky valve, or interfere with the free working of the valve spindle in its guide. With reference to the spindle guide, it should be added that the spindle should not be an exact fit in the guide, as lubrication of these surfaces is difficult, and the expansion considerable, giving rise to a tendency of the spindle to stick, so that a clearance of about 1 mm. (0·04 of an inch) should be left. It is better to ensure true vertical movement by making the spring plunger a good fit.

The valves are always of steel, except those for the exhaust, which often have cast-iron heads, the better to withstand the temperature (Fig. 173). The cages are almost always of cast iron.

Inlet and scavenging valves, cooled as they are by the current of air which passes them, do not require any special provision to avoid excessive heating (Figs. 171, 172, and 175). But in the case of those for the exhaust, which are subject every cycle to the burning of the exhaust gases, water cooling of the cage is necessary (Fig 173), except with diameters less than

* Only in engines of the lowest power, where the smallness of the head permits of ready removal, are the guide and the valve seating machined in the head itself.

60 to 70 mm. ($2\frac{1}{4}$ to $2\frac{3}{4}$ inches). For larger valves the valve heads and spindles are also water-cooled (Fig. 174).

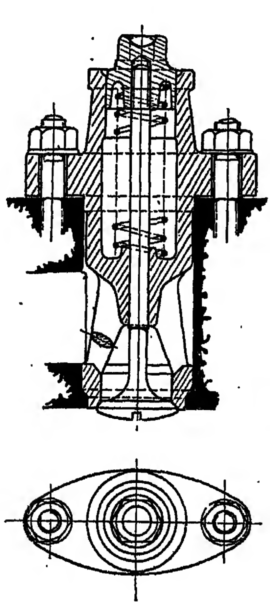


Fig. 171.

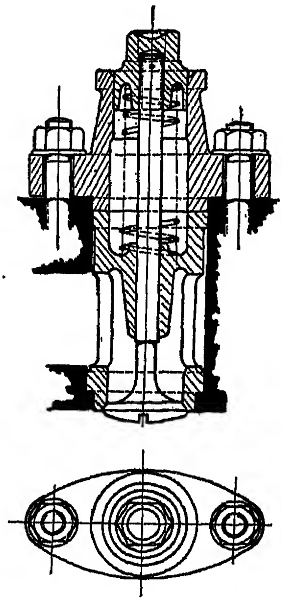


Fig. 172.

Figs. 171 and 172.—Types of Suction and Scavenging Valves.

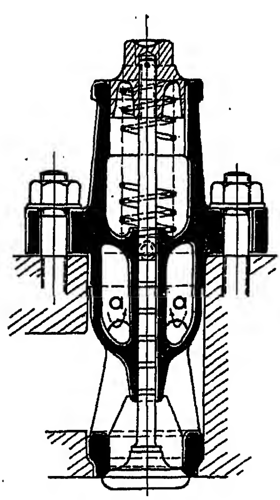


Fig. 173.—Exhaust Valve with Cast-iron Head.

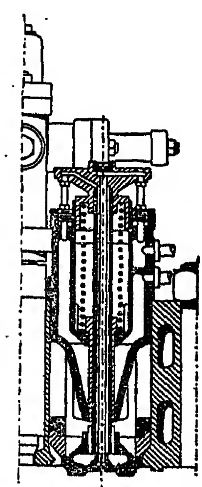


Fig. 174.—Water-cooled Exhaust Valve.

Fig. 175 shows the scavenging valve of a small marine Sulzer engine, the removable casing cover of which is not shown. This is a very common construction, and confines the deformation consequent upon tightening up the nuts to the casing cover. Other types of construction of cages and valves may be seen in the plates throughout the book.

With four-cycle engines, the mean velocity of the air and of the exhaust gases past the valve seat is generally from 30 to 40 metres (98 to 132 feet) per second,* determined by the valve diameters, which are made as large as the design of the head will permit, and by the lift, which is always about one-fourth of the smaller diameter of the seating.

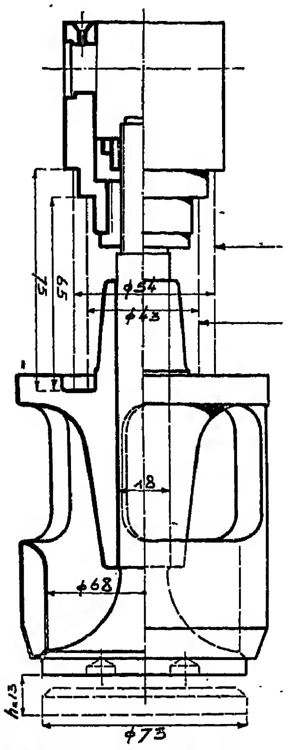


Fig. 175.—Scavenging Valve and Cage of Small Marine Sulzer Engine.

As already stated, in order to provide valves of the largest possible diameter, small recesses are formed in the cylinder walls in way of the valves, and sometimes also the fuel injection valve is placed out of the centre of the head.

It is difficult to calculate the diameters of scavenging valves for two-cycle engines, since the diameter and the lift vary with the pressure of the scavenging air, the speed of the engine, the time available for scavenging, the method adopted for scavenging, and, above all, the available space.

Valve springs may be calculated for a load of 0·5 to 0·6 kg. per sq. cm. (7 to 8·5 lbs. per square inch) on the area of the valve; for high-speed engines the accelerating force, which the spring gives, should be sufficient to keep the valve spindle in contact with its operating mechanism when the valve is closing. The method for this calculation is explained in the following chapter.

Fig. 176 represents a starting air valve; the flange is often separate, and holds the cage against its seating machined in the cylinder head. A joint at *g* maintains tightness against external leakage. The air enters from the passage *e*, and fills the whole of the valve chamber and the interior of the cage.

The air pressure tending to open the valve is balanced by the large portion of the spindle at *c*, in order to avoid the necessity of an

* By the mean velocity is understood that obtained from the formula $A \cdot V = a \cdot v$, in which A and V are the area and the mean velocity of the piston, v the velocity of the gas, and a the area through the valve opening. If h is the lift of the valve and d its smallest diameter, $a = \pi \cdot d \cdot h$, and since h is never much less than $\frac{d}{4}$, it may be said that $a =$ about $\frac{\pi}{4} d^2$, and so

$$v = \text{about } \frac{4\,A \cdot V}{\pi\, d^2}.$$

excessively strong spring to keep the valve shut. It is desirable, however, to provide a fairly strong spring for this valve, since any appreciable leakage during the compression stroke may make it impossible to start the engine.

The part *c* of the valve spindle is frequently provided with small spring rings similar to those used with the main piston.

The starting air valve of a Sulzer marine engine is shown in Fig. 177.

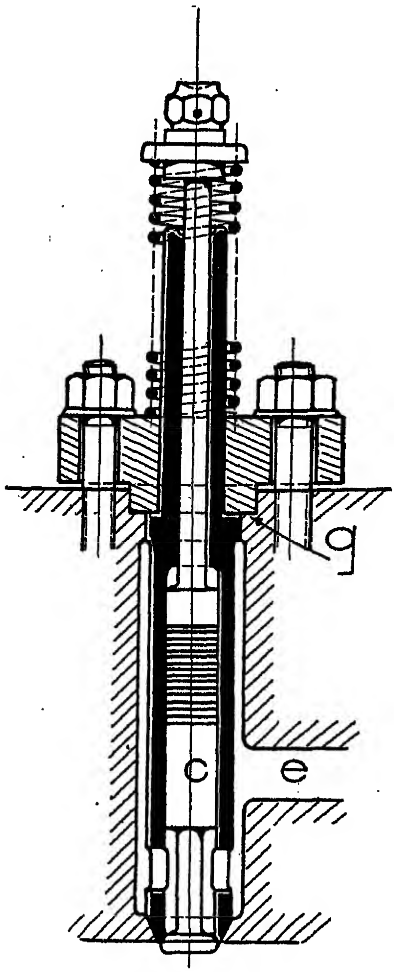


Fig. 176.—Typical Starting Air Valve.

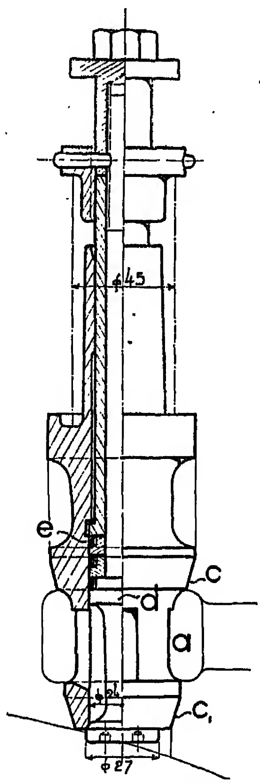


Fig. 177.—Starting Air Valve of Sulzer Marine Engine.

he compressed air enters the chamber *a*; *c* and c_1 are the two conical seatings ısuring external and internal tightness respectively. The spring rings *e* ırried by the enlargement *d* of the valve spindle serve for the usual balance pressures.

With large M.A.N. horizontal engines, the starting air valve is actuated by a small compressed air piston instead of the usual lever mechanism. The starting air enters through the passage *a* (Fig. 178).* The air for acting

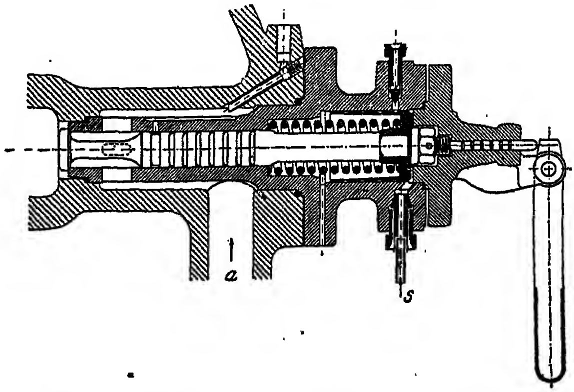


Fig. 178.—Starting Air Valve of M.A.N. Horizontal Engine.

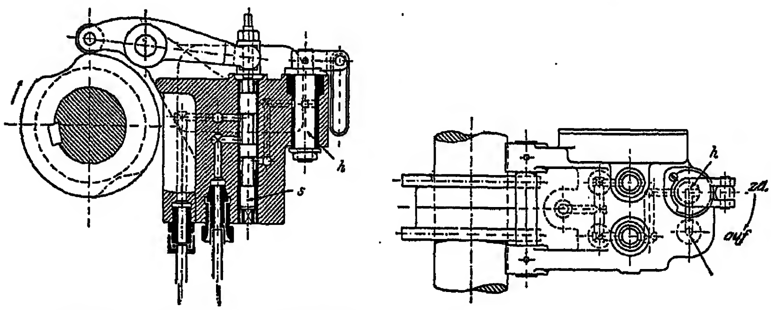
on the small actuating piston is controlled by a small distributor, and enters by the pipe *s*.†

Fuel Injection Valves.—The fuel injection valve is one of the characteristic parts of the Diesel engine. Its functions are twofold: firstly, that of a valve to introduce the fuel oil into the cylinder at the correct moment; and, secondly, that of a sprayer to divide the fuel into minute particles.

The valve proper is formed by the needle *b* (Fig. 181), ending in a cone, held on a conical seating by the valve spring (Figs. 181 and 184). The lever

* *Zeitsch. des Ver. deut. Ing.*, 1911, p. 1313, *et seq.*

† The distributor is a cylindrical valve *s* (Figs. 179 and 180) actuated by a cam. The cock *h* serves to throw the distributor in and out of gear by opening or cutting off the air supply. These sketches show two distributors and two cams as applied to a two-cylinder horizontal engine.



Figs. 179 and 180.—Actuating Mechanism of Starting Air Valve of M.A.N. Horizontal Engine.

c, actuated by a cam, raises the needle at the required moment, establishing communication between the fuel injection valve casing and the cylinder.

Spraying is effected by means of a number of washers (Figs. 183 and 184), provided with small holes, and by a cone grooved along its generating lines, threaded on a tube *g* also of bronze (Fig. 181).*

The steel diaphragm O (Figs. 181 and 184), having a small hole at the centre, plays a certain part in the spraying action.

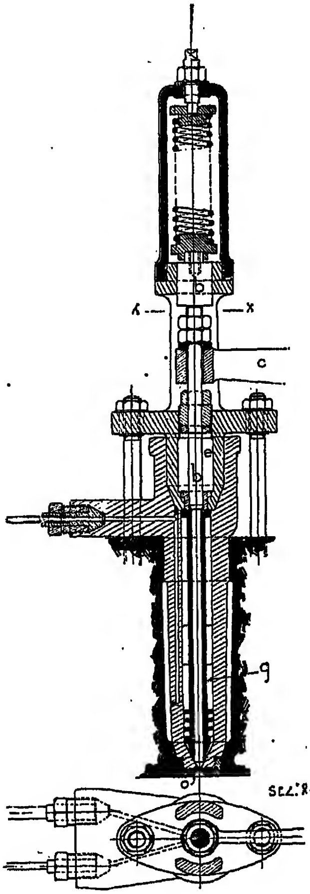


Fig. 181.

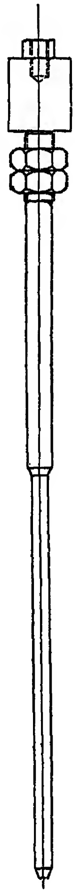


Fig. 182.

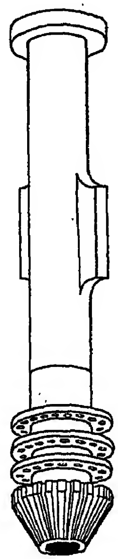


Fig. 183.

Figs. 181 to 183.—Fuel Injection Valve.

The oil, delivered under pressure from the fuel injection pump, is led into the fuel injection valve just above the perforated washers, and high-pressure compressed air from the fuel injection bottles fills the annulus

* In some Sulzer fuel injection valves the tube is of bronze, but the washers and the cones are of cast iron.

surrounding the sleeve *g*. When the valve spindle lifts, the air at 50 to 70 atmospheres—700 to 1,000 lbs. per square inch—rushes into the cylinder, in which the pressure is 30 to 35 atmospheres—430 to 500 lbs. per square inch—drawing with it the oil, which, in passing through the holes in the washers, the channels in the cone (causing the particles to become greatly agitated), and the orifice washer *o*, is divided into a fine mist. The fuel oil in this divided state enters the hot combustion chamber of the cylinder at the end of the compression stroke, and spontaneously ignites. The combustion takes place practically at constant pressure throughout the period of the stroke during which the oil continues to be forced into the cylinder.

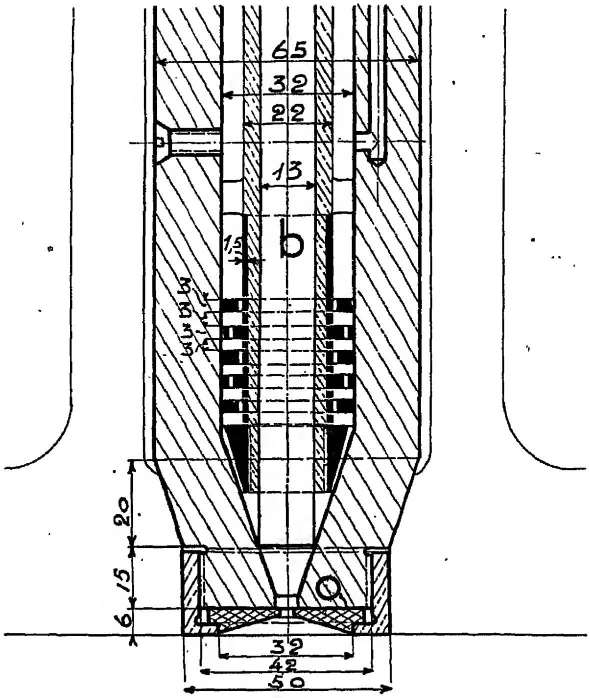


Fig. 184.—Sprayer of Fuel Injection Valve.

The valve actuating gear is generally arranged in such a way that the lift and time of opening of the valve are not variable with the load on the engine, on account of which, if the difference in pressure between the interior of the fuel injection valve casing and the cylinder remains constant, the velocity of efflux and the quantity of air issuing with each working stroke do not vary, irrespective of the power the engine may be developing.

On the other hand, the quantity of oil delivered by the fuel injection pump, and therefore the quantity of oil which finds its way into the fuel injection valve for each combustion stroke does vary with the load. It

will be seen that if the velocity of the efflux of the air is constant, the velocity with which the oil is drawn by it into the cylinder will also be constant, whatever may be the quantity delivered by the fuel injection pump as determined by the load on the engine, but the varying quantity of fuel will require different periods of time for its injection.

It is for this reason that in the indicator diagrams of a Diesel engine the load influences only the duration of the period of combustion, in exactly the same manner as it would influence the period of admission with a steam engine having variable expansion.

It has been seen (p. 57) that the thermal efficiency of the constant-pressure cycle increases as the period of fuel injection decreases, which is perfectly rational; the only objection which can be raised is that at very low powers there is an appreciable loss of fuel injection air. In fact, since the time of opening of the valve is constant, whilst that of the injection of the oil is variable, the air continues, after the fuel oil contained in the casing has been injected, to expand uselessly into the cylinder until the valve has closed.

Some engines of high power are provided with a mechanism varying the lift of the fuel injection valve (Fig. 219, p. 153). Otherwise, as an expedient for obviating the loss of fuel injection air above mentioned, its pressure is reduced when the engine is not developing its full power.

The fuel injection valve is composed of a cast-iron casing in which is screwed a bronze tube to carry the perforated washers and the atomising cone. The casing is kept on its conical seat by a casting secured to the cylinder head by two or four studs of large size. In this upper casting or casing cover is carried the stuffing-box *e* for the valve *b* (Fig. 181), and this casing cover is connected to the valve spring box by means of arms between which the valve lever *c* works.

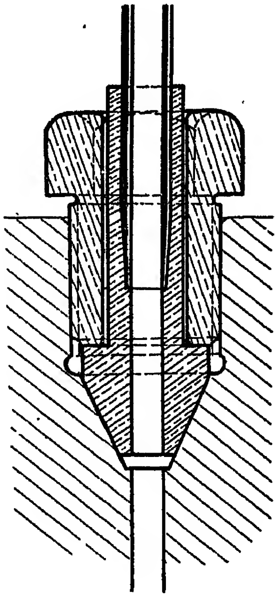


Fig. 185.—Fuel or Air Pipe Connection to Fuel Injection Valve.

The pressure of the spring is regulated by a screw, which permits the tension being taken off the spring before dismantling. The check nut on the valve serves as a seating for the lever *c*, and regulates the clearance between the cam and the roller when the valve is in the closed position.

The stuffing-box is formed by a nut and a neck bush, between which packing of well hammered lead or white metal shreads is generally provided. A little graphite mixed with the shreds of the metal gives the requisite properties of softness and lubrication.

The fuel and the compressed air are led to the fuel injection valve by two copper or solid drawn steel pipes (that for the air slightly larger than that for the fuel), which are attached to the casing by means of bronze unions (Fig. 185).

The air enters the casing directly at the top, and the fuel, by means of small holes drilled in the metal, is led to just above the perforated washers.

The drilling of these long and small holes presents certain difficulties, to overcome which recourse is sometimes had to other expedients, such as leading the fuel into an annulus formed between the internal wall of the casing and a copper sleeve provided at the bottom with small holes.

The fuel injection valve described corresponds generally to the type

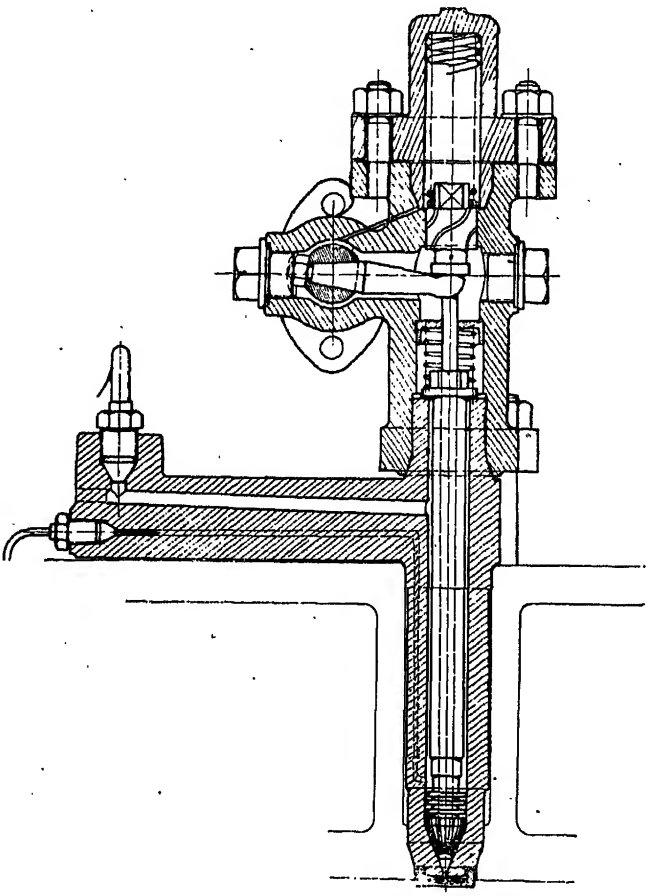


Fig. 186.—Section through Sulzer Type of Fuel Injection Valve.

used by almost all the notable constructors of Diesel oil engines (M.A.N., L.W., Tosi, Grazer, etc.). A construction differing, however, from the one described is that of the Sulzer type and its derivatives, illustrated in Figs. 186, 187, and 188. In this case, the valve lever does not act directly on the valve spindle, but indirectly through the medium of a horizontal shaft, the part *a* of which is reduced in diameter, and fixed at one end, acting in this

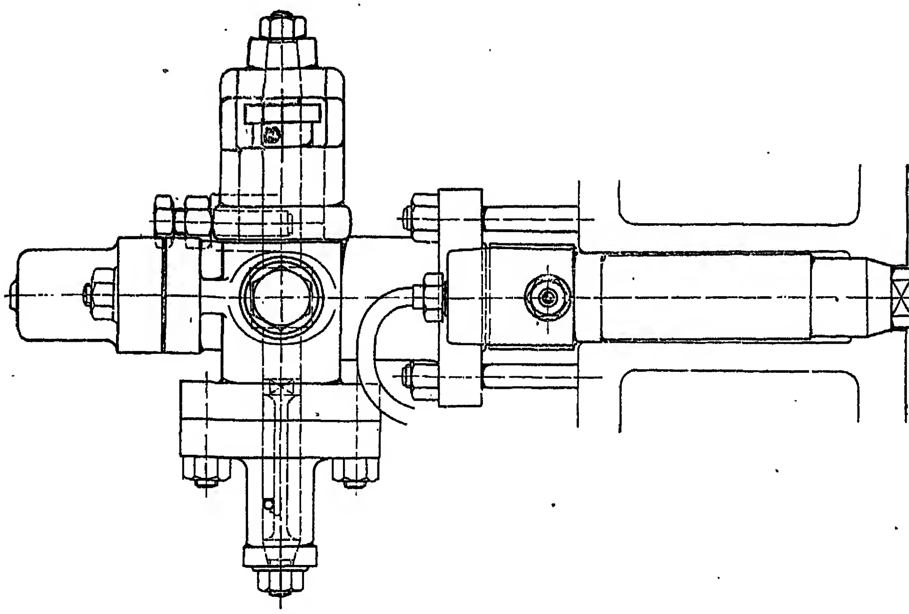


Fig. 188.—Front Elevation of Sulzer Type of Fuel

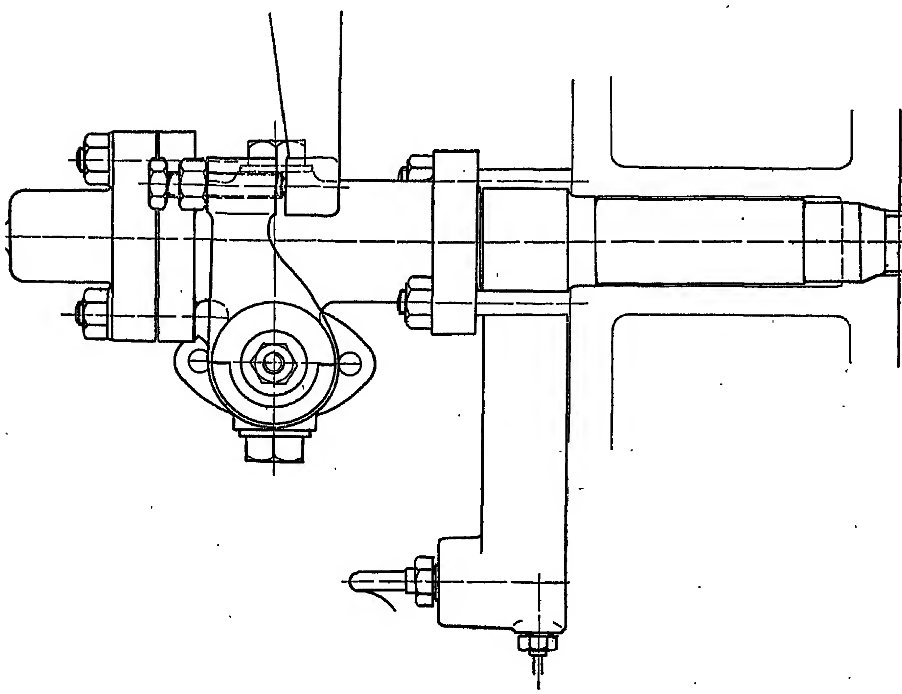


Fig 187.—External View of Sulzer Type of Fuel Injection

way as a torsion spring (Fig. 188). For tightness, a stuffing-box *e* is provided.

The valve lever is carried thus to one side of the valve casing, and as shown in Figs. 187 and 188, is so constructed as to permit of the withdrawal of the fuel injection valve without interfering with the valve lever.

In the following chapter the manner in which other constructors have achieved the same result is exemplified.

The Hesselman (Polar-Benz) fuel injection valve, part of which is shown diagrammatically in Fig. 189, is quite different from the ordinary type.

The main characteristic is that one special part serves instead of the perforated pulverising washers and the grooved cone of the usual type. This special part surrounds the spindle with a little clearance, which is increased at the bottom to form a species of ejector *a*, through which the fuel injection air passes (as shown by dotted arrows in Fig. 189).

The fuel oil is led as usual to the bottom by means of holes drilled in the

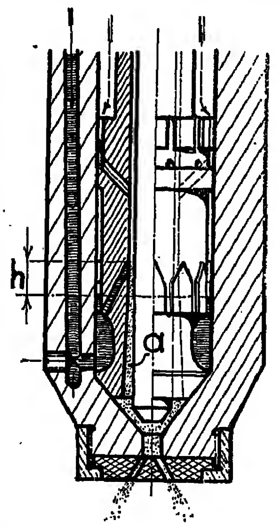


Fig. 189.—Hesselman (Polar-Benz) Fuel Injection Valve.

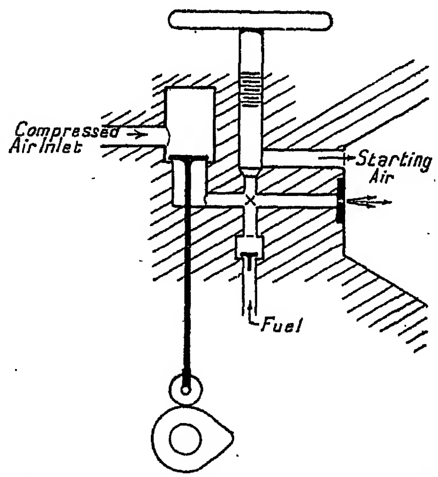


Fig. 190.—Diagram of Lietzenmeyer Type of Fuel Injection Valve.

cast-iron casing, and when the valve is shut the oil reaches a certain level, depending only upon the quantity delivered by the pump. When the valve lifts, since the pressure of the air acts on the upper surface of the fuel oil, while the pressure in the ejector is reduced due to the air entering the cylinder, the oil rises in the narrow oblique holes and flows into the ejector *a*, where the current of air draws it into the cylinder. If the drop of pressure produced by the ejector is sufficient to produce a given maximum head *h*, measured from the level of the oil in the annulus between the atomiser and the casing and that in the small oblique passage, then, when the column of oil in the external annulus is lower than given by height *h* (see Fig. 189), the fuel automatically ceases to be drawn into the cylinder, and the combustion period is finished. On this account, the level of the oil in the casing varies

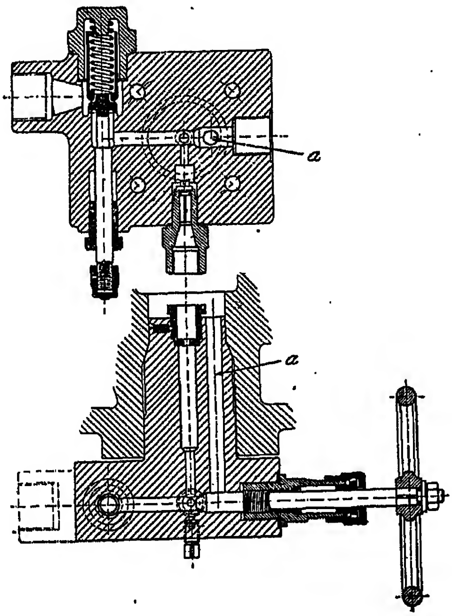
with the quantity of oil delivered by the pump, but, at the end of the combustion period, is always lower, by the height *h*, than the upper orifices of the oblique holes.

This fuel injection valve has the advantage over the usual type that regulation of the pressure of the fuel injection air with the variation of the load is not required, but the design does not lend itself to the easy adjustment that in other types is obtained by varying the number of the pulverising washers and the diameter of the holes in them.

Another interesting and original design is that applied to horizontal engines of the Lietzenmeyer and similar types (Dingler, Körting, etc.). This design has been evolved to remove from the fuel injection pump the necessity of working against the pressure of the fuel injection air, to diminish in this way the power thereby absorbed, as well as leaks, and the necessity for very well-made joints and carefully packed glands, etc.

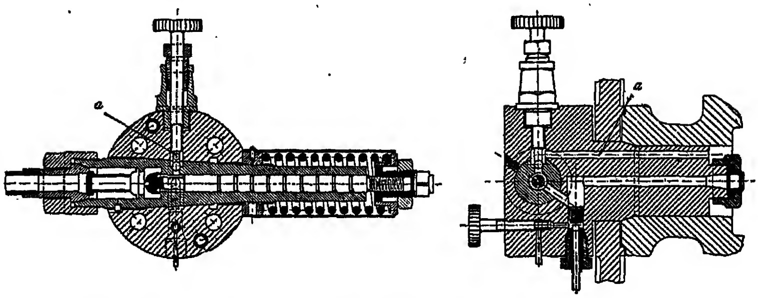
This fuel injection valve is so disposed (Fig. 190) that, instead of opening or closing the communication between the valve chamber and the cylinder, it opens or closes the conducting passage for the injection air, whilst the pulverising chamber is always subject to the pressure of the cylinder, with which it is in direct communication. It is sufficient then that the fuel injection pump should deliver during the suction and exhaust strokes of the engine, in order that the pressure against which it works may be a negligible quantity.

When at the correct moment the valve opens, the influx of air draws the quantity of oil, previously delivered by the pump, through the pulverising media into the cylinder in the usual finely divided state.

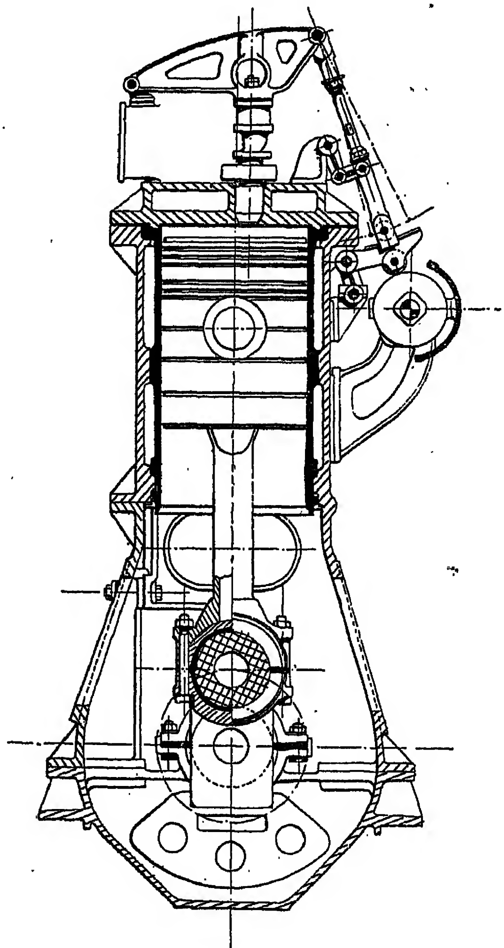


Figs. 191 and 192.—Constructive Features of Lietzenmeyer Fuel Injection Valve.

This valve serves also as a starting air valve in the following way:—The cam which serves to actuate the injection valve is doubled—*i.e.*, alongside the nose of the injection cam is another for starting the engine, capable of lifting the valve with a slightly retarded opening and for a slightly longer time. The roller of the fuel injection valve push rod may be displaced axially, so as to roll on one or other of these two cams. When it is desired to start the engine, all that is necessary is to place the roller on the second cam, and to open a stop valve permitting communication with the cylinder through a passage of larger diameter than that of the injection washer of the fuel sprayer.



Figs. 193 and 194.—Details of Dingler Type Fuel Injection Valve.



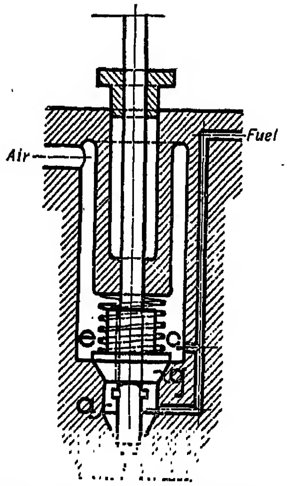


Fig. 195.—Diagram of Sabathé Fuel Injection Valve.

Fig. 196.—Arrangement of Variable Lift Valve Gear (Sabathé).

The constructive features of a fuel injection valve of this type (Lietzenmeyer) are to be seen in Figs. 191 and 192; Figs. 193 and 194 show the Dingler type.*

A special fuel injection valve is used in Sabathé engines, which work on the mixed combustion cycle (Fig. 56, p. 47).

The fuel is injected in two phases; the first, when the piston is at the top dead centre, produces combustion almost at constant volume, and the second, entering later for the second phase, gives combustion at constant pressure. At reduced powers the part of the diagram at constant pressure is greatly shortened, as in Diesel engines, or is even entirely eliminated, in which case the cycle is that of an explosion engine. In Fig. 195 is shown the diagrammatic arrangement of a fuel injection valve of this type. Compressed air fills the spaces *e*, and also *a*, through the channel provided in the guide of the valve *g*. The fuel is led into the lower space *a*, which is insufficient to contain all, and it overflows by means of the hole *c*, into the annulus *e*; when the valve commences to lift, first, the fuel contained in *a* is driven into the cylinder, producing thus the phase of combustion characterised as that at constant volume; then the valve continuing to rise lifts, by means of two projections with which it is provided, the second valve *g* permitting the remaining fuel contained in the annulus *e* to enter the cylinder and to produce the phase of combustion at constant pressure. When the power varies, and with it the delivery of the pump, the quantity of fuel which overflows through the hole is also varied, and whilst the phase of combustion at constant volume remains unchanged, that at constant pressure is varied. At low powers the quantity of fuel is insufficient to reach the level of hole *c*, and combustion at constant pressure is entirely eliminated.

The general arrangement of the valve actuating gear is seen in Fig. 196, by means of which at reduced loads the lift of the valve itself is varied, and at very light loads the valve *g* (Fig. 195) is not lifted off its seat, thus giving economy of compressed air. This latter consideration is of considerable importance, as this engine is designed for special application to marine propulsion where a reduction of load means a reduction of revolutions—*i.e.*, an increase of the time during which the fuel injection valve is open, and a consequent large augmentation of the quantity of compressed air used.

* *Zeitsch. des Ver. deut. Ing.*, 1911, pp. 1313, *et seq.*

CHAPTER X.

VALVE ACTUATING GEARS.

In this chapter the valve actuating gears of Diesel engines will be considered, dealing only with the types suitable for engines which rotate in one direction.

In this category all land engines are included, and, as will be seen, certain marine engines where special means are provided, either in the propeller shaft or embodied in the propeller itself, to reverse the direction of motion of the ship without altering that of the main engines.

Some of the best solutions of well-known constructors to solve the problem of the reversal of marine engines will be given in the following chapter.

The valves of Diesel engines are always positively operated, generally

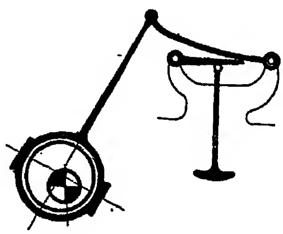


Fig. 197.—Diagram of Eccentric and Rolling Lever Valve Actuating Gear.

by a lever actuated by cams and in some cases by eccentrics (Fig. 197), in which case an arrangement of rolling levers to increase the velocity of lift and of closing the valve is necessary.* The eccentric drive is more expensive than that by means of cams, and is rarely met with.

The diagram (Fig. 198) shows the general arrangement of the valve actuating gear for most four-cycle Diesel engines, in which the cam shaft e revolving at half the speed of the engine crank shaft carries the cams $a_1 b_1 c_1 d_1$

* For calculations relating to rolling levers, see H. Dubbel, *Grossgasmaschinen*, Berlin, 1910, p. 69, *et seq.*, and also Holzer, *Wälzhebel*, *Zeitsch. des Ver. deut. Ing.*, 1908, p. 2043, *et seq.*

acting upon the levers *a b c d* of the suction, fuel injection, starting air and exhaust valves respectively. The fixed shaft *g g* attached to the cylinder head by two pillars (see Fig. 152, p. 118) acts as a fulcrum for these levers. Levers *a* and *d* are mounted directly on the shaft *g g*, whilst *b* and *c*, for the fuel injection and starting air valves respectively, pivot on an eccentric sleeve, working in its turn upon the shaft *g g*, and provided with a lever *o*, of which the function will be explained later.

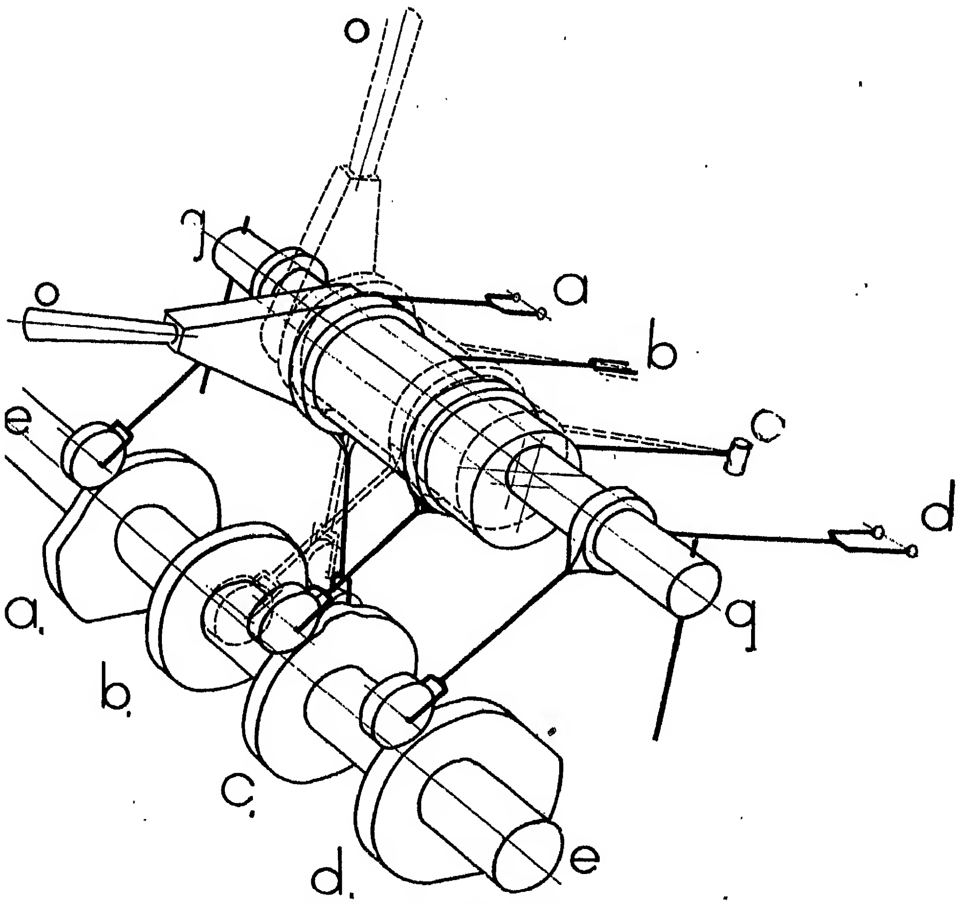
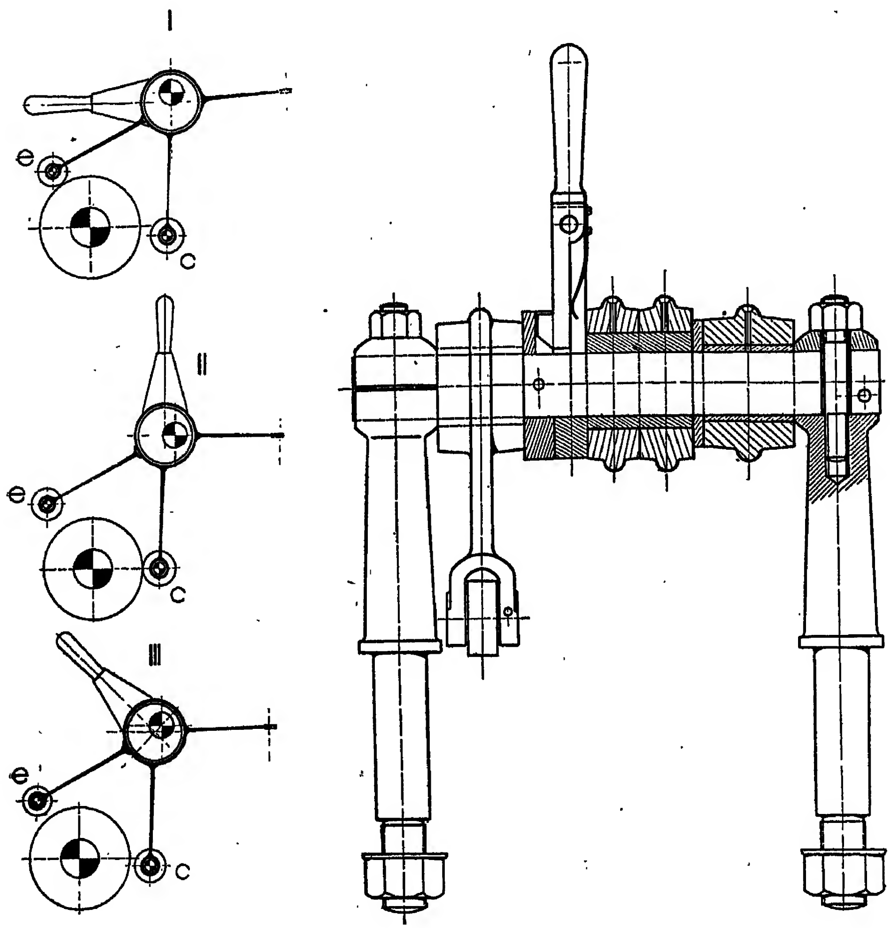


Fig. 198.—Diagram of Valve Actuating Gear for Four-cycle Engine.

The levers *a*, *c*, and *d* have rollers towards the outside of the cam shaft, so that the cams in acting upon them open the valves downwards, whilst the lever *b* of the fuel injection valve has its roller inside—*i.e.*, between the cam shaft and the cylinder—so that the cam in this case may lift the valve. The design of the various valves discussed in Chapter IX. necessitates this particular arrangement of drive.

The function of the handle and of the eccentric sleeve, upon which the levers *b* and *c* are mounted, is clearly shown in Figs. 199 to 201, in which *e* is the starting air valve roller and *c* that for the fuel injection valve. When the handle is in the horizontal position (I) the eccentric sleeve to which it is connected removes the roller *c* of the fuel injection lever away from its



Figs. 199 to 201. Fig. 202.

Figs. 199 to 202.—Function of Cam and Eccentric Sleeve for Starting Air and Fuel Injection Valves.

cam, and causes the roller *e* of the starting air valve to approach its cam by the same amount.

The displacements are such that the cam does not act on the roller *c*,

and the fuel injection valve does not lift, whilst, on the other hand, the starting air valve is in action. When the handle is in this position, the engine will start to run on compressed air as soon as the starting air bottles are put in connection with the starting air valve. By moving the lever to the vertical position (II), exactly the opposite occurs—*i.e.*, the fuel injection valve is brought into action; whilst that for starting is thrown out of gear, since its roller is lifted from the cam to prevent contact taking place. This position corresponds to that of normal running.

In the intermediate position of the lever (III) neither of the two valves are in action, and the engine will stop.

The general arrangement of this very simple mechanism for manœuvring, as usually applied, is seen in Fig. 202, which also shows the arrangement of the fulcrum shaft. In some cases, to avoid the necessity of ascending to the top platform to manœuvre the engine, the handle is brought to a lower position on the engine, and acts on the sleeve through tie-rods, and in certain multi-cylinder engines provision is made to permit of one lever operating at the same time the valve gears of all the cylinders. This provision—necessary with marine engines—makes the operation of starting less troublesome, although in land installations rapidity of manœuvring is not of great consequence, and it is often preferable to put the cylinders into action one at a time, commencing with the one which experience has shown to be the readiest starter, whilst the others continue to be run by compressed air.

For land engines many constructors, to effect economy of production, provide starting gear on only two or even on one cylinder with engines of three or four cylinders.

The valve actuating gears of two-cycle engines depend to some extent upon the method of scavenging. The exhaust always takes place through the exhaust ports cast in the cylinder, and the regulation of the time of operation of the exhaust is dependent upon the piston travel, so that there are no exhaust valves in the cylinder head.

If the scavenging air is also introduced by ports, the valve actuating gear proper is confined to the starting air and the fuel injection valves.

On the other hand, with cylinder head valve scavenging by means of one, two, or four such valves, as explained on p. 122, these valves are always simultaneously opened and shut, and the general arrangement of the valve gear is frequently similar to that shown in Fig. 203.

The cam shaft *e e* revolving at the same velocity as the crank shaft, carries the cams c_1 and b_1 for the starting air and fuel injection valves, and the two cams d_1 d_1 identical in every way with one another, and keyed on at the same angle as each other, for the scavenging valves.

The scavenging valve levers at *r r* act upon the front pair of valves, and by means of hooks *d d* connect with two other levers moving about the fixed pins $g_1 g_1$, which in their turn operate the back pair of valves at $r_1 r_1$. Since the distance between the points *d* and the fulcrums *g* and g_1 are equal, and the distance *d r* is equal to $d\, r_1$, the lift of the four valves will be the same.

For equal lifts giving the same period of opening of the four valves, besides the arrangement described, which is shown in its application in

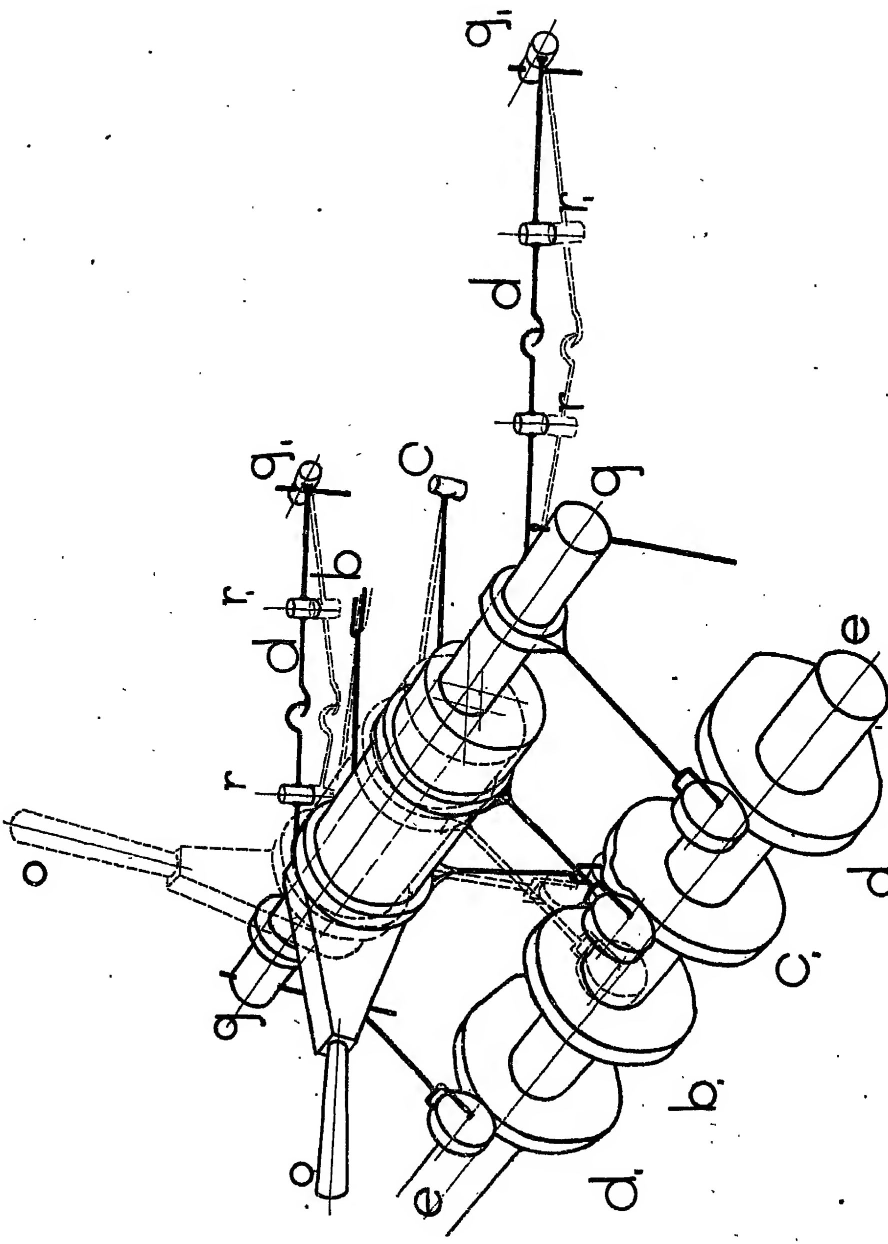


Fig. 203.—Diagram of Valve Actuating Gear for Two-cycle Engine with Four Scavenging Valves in Cylinder Head.

Fig. 204 (Carels), many others may be devised, including that of four levers and four independent cams.

In Fig. 205 the Sulzer valve actuating gear for two-cycle engines is shown. There are four scavenging valves (the two shown in the figure are indicated by *a*), whilst *b* is the fuel injection valve and *c* the starting air valve. The four scavenging air valves are actuated in pairs by two levers operated by

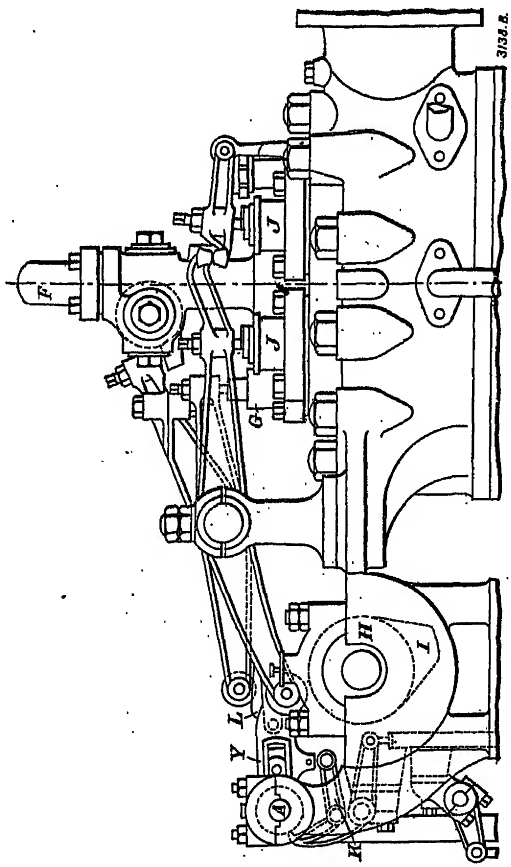


Fig. 204.—Valve Actuating Gear of a Carels Two-cycle Engine with Four Scavenging Valves in Cylinder Head.

two identical cams keyed on to the cam shaft at the same angle; *e* is the usual fulcrum shaft and *g* a fulcrum pin, placed so that the distance from *g* to *e* is equal to that between the pins of the crosshead *d*. This system of two shafts with the crosshead *d*, the actuating lever and the link *l*, forms a parallelogram, which ensures equal lifts and equal openings for the couple of valves *a a*; the shaft *g* serves also as a fulcrum shaft for the starting air valve lever.

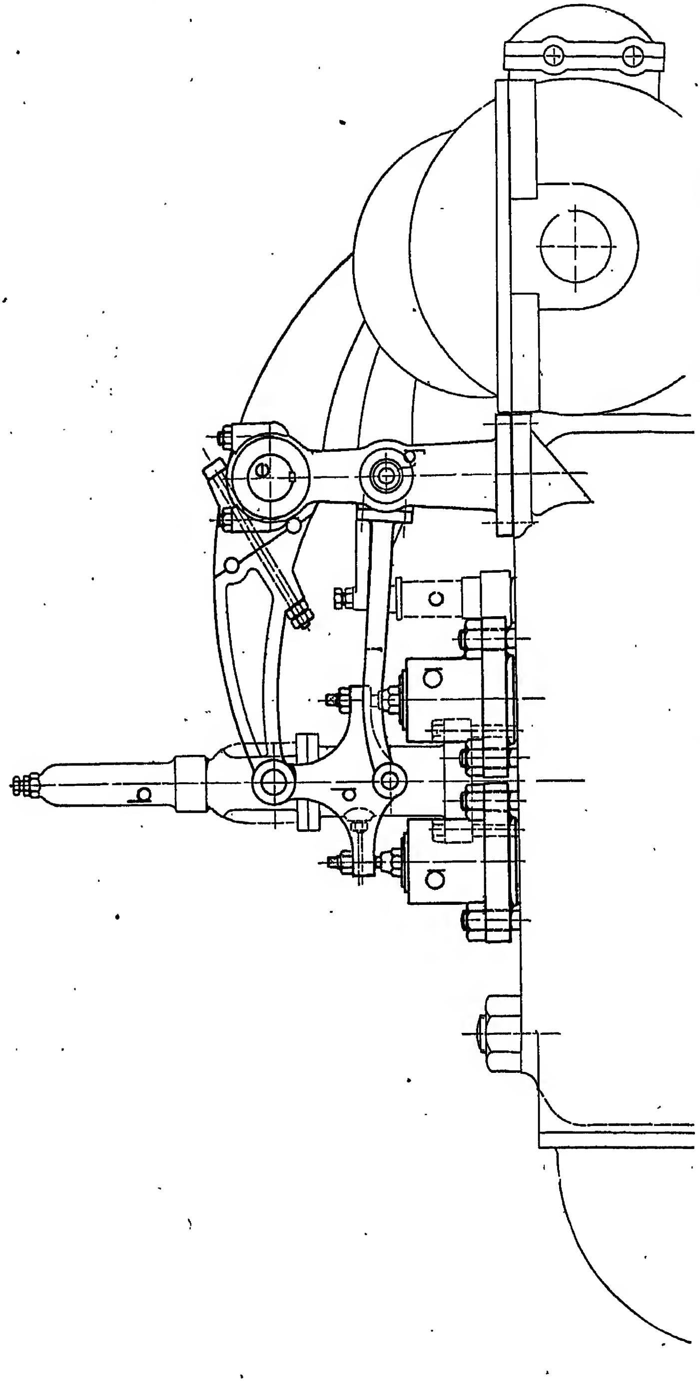
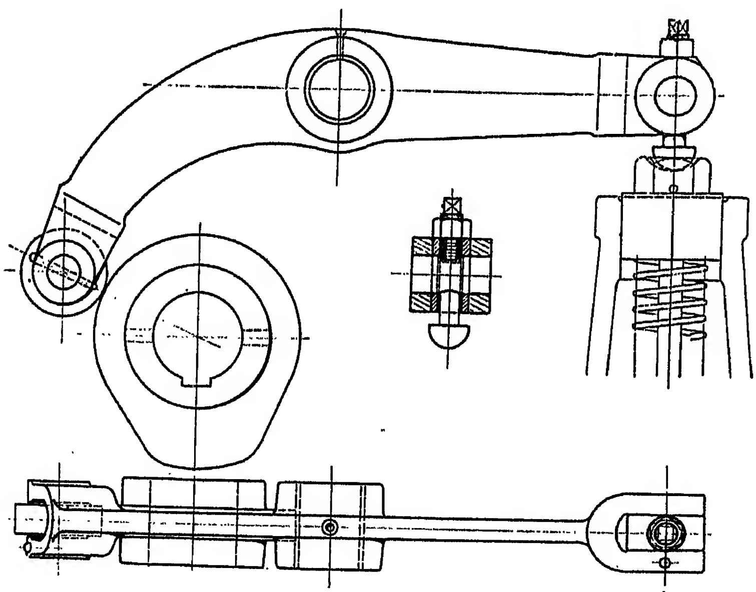


Fig. 205.—Valve Actuating Gear of a Sulzer Two-cycle Engine with Four Scavenging Valves in Cylinder Head.

Valve Levers.—From the constructive point of view, the cams and the levers for the inlet, exhaust or scavenging valves are generally similar to those shown in Figs. 206 and 207. A roller of hardened steel rotates on the nose of the cam and causes the lever to turn about its fulcrum. The spring which holds the valve on its seat also causes the roller to remain in contact with its cam during the periods of opening and closing of the valve. The lever acts upon the valve by means of an adjustable pin, which admits of the clearance being varied between the lever roller and the circular part of the cam when the valve is shut. This clearance is generally between 0·5 and 1 mm. ($\frac{2}{100}$ and $\frac{4}{100}$ inch).

In some engines the valve levers are not directly acted upon by the cams, but receive their motion through push-rods, on the extremities of which the



Figs. 206 and 207.—Type of Lever and Cam suitable for either Suction, Exhaust, or Scavenging Valve.

rollers are mounted (Fig. 208). This arrangement is adopted generally to permit of the cam shaft being in a lower position on the engine.

With high-speed engines, especially those working on the two-stroke cycle, every effort is made to reduce the dimensions of the lever to the minimum. Long levers are necessarily heavy, and if the opening of the valve is rapid, recourse must be had to an extra strong spring, to give, during the period of shutting, sufficient acceleration of the valve to maintain contact between the cam and its roller. Moreover, with high-speed engines, long levers are apt to cause noisy working, with a greater chance of breakdown. Rolling levers (Fig. 197, p. 140, and Figs. 245, 246, p. 173, Fiat) are

adopted primarily for reasons of silence, and in some cases (Fig. 209, Carels) the levers are totally eliminated and the cams act directly upon the valves. To remove the valve for inspection, it is usually necessary first to dismantle the fulcrum shaft with all its levers; but in Fig. 205 a design to permit of dismantling the valves without this extra work being necessary is clearly shown, and renders comparatively simple an operation that is complicated and of fairly frequent occurrence, and one requiring great care, especially with very large engines.

The levers are divided and held together by a bolt, whilst two circular keys ensure exact registration of the two parts. To dismantle the valve, it is sufficient to slack back the bolt, and remove the horizontal part of the lever. A similar system is adopted by Tosi for large engines.

The valves which it is necessary to examine most frequently are those

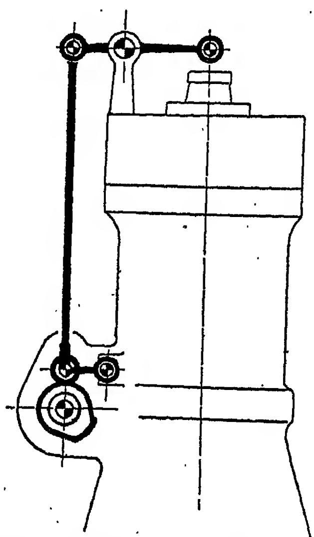


Fig. 208.—Diagram of Valve Lever indirectly operated through a Push Rod.

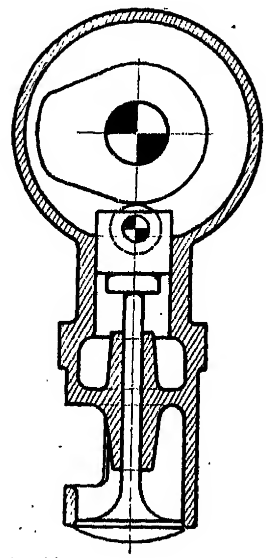


Fig. 209.—Scavenging Valve directly operated by Cam.

for fuel injection and starting air, and for these some system to permit of ready dismantling without interfering with the remainder of the valve gear is most urgently required. Besides being fitted to large engines, arrangements satisfying this condition are applied to small engines, and have the further advantage of permitting the variations of the clearances between the cam and its roller being more conveniently made than with a nut and check nut on the fuel injection valve spindle, as illustrated in Fig. 181, p. 131. Such a system has been described in connection with the Sulzer fuel injection valves (Figs. 186, 187, and 188, pp. 134 and 135).

A different design, adopted also by Messrs. Sulzer, is illustrated in Fig. 210, consisting of a split lever similar to that for the scavenging valves (Fig.

205), in which a pin with a locking nut is substituted for one of the keys, and serves conveniently to adjust the clearance between the roller and the cam.

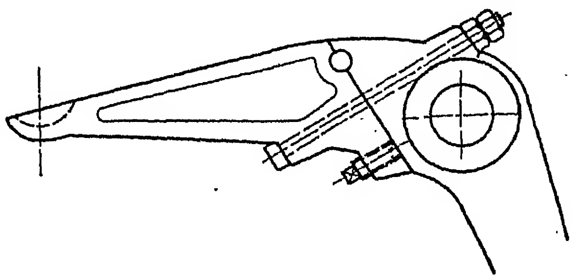


Fig. 210.—Sulzer Type of Valve Actuating Lever giving easy Access to Valves for Examination.

One of the earliest and most satisfactory constructions is that of Langen & Wolf (Fig. 211), where the pin *c* only requires to be removed to permit of

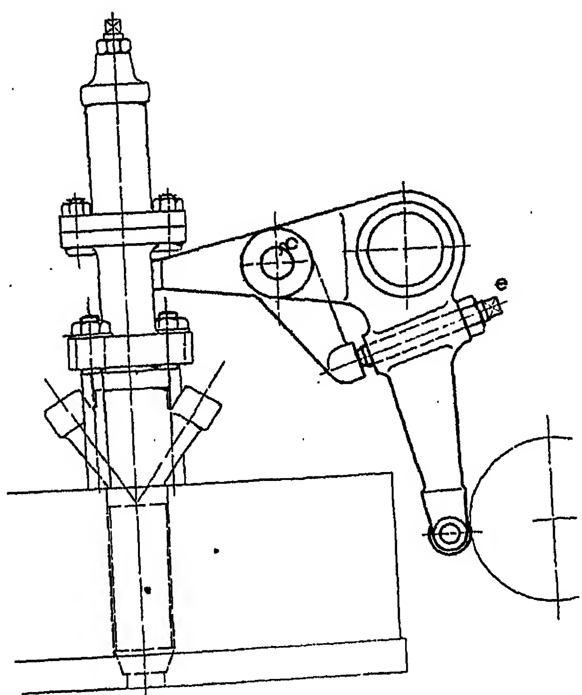


Fig. 211.—Fuel Injection Valve Actuating Lever (Langen & Wolf).

the dismantling of the fuel injection valve, and the set screw *e* serves to adjust the clearance between the roller and the cam.

The arrangement shown in Fig. 212 has been recently adopted for Tosi engines.

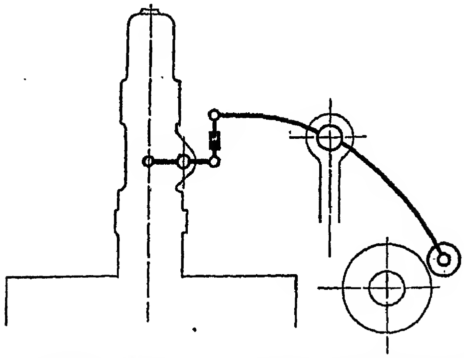


Fig. 212.—Diagram of Tosi Fuel Injection Valve Actuating Gear.

Cam Shaft and Bearings.—With four-cycle engines the cam shaft is generally driven by gear wheels of the helical type, which reduce

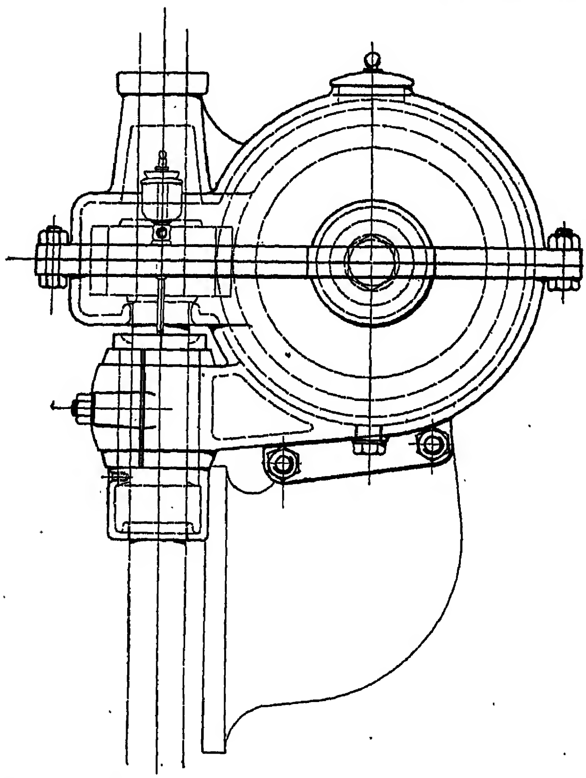


Fig. 213.—Arrangement of Spiral Gearing for driving Cam Shaft.

the number of revolutions of the shaft to one-half that of the engine crank shaft.

With two-cycle engines the speed of revolution of the cam shaft is the same as that of the engine crank shaft. The gear wheels are enclosed (Fig. 213) in a casing containing the lubricating oil and carrying a bearing for the intermediate vertical shaft, which latter runs at the same speed of revolution

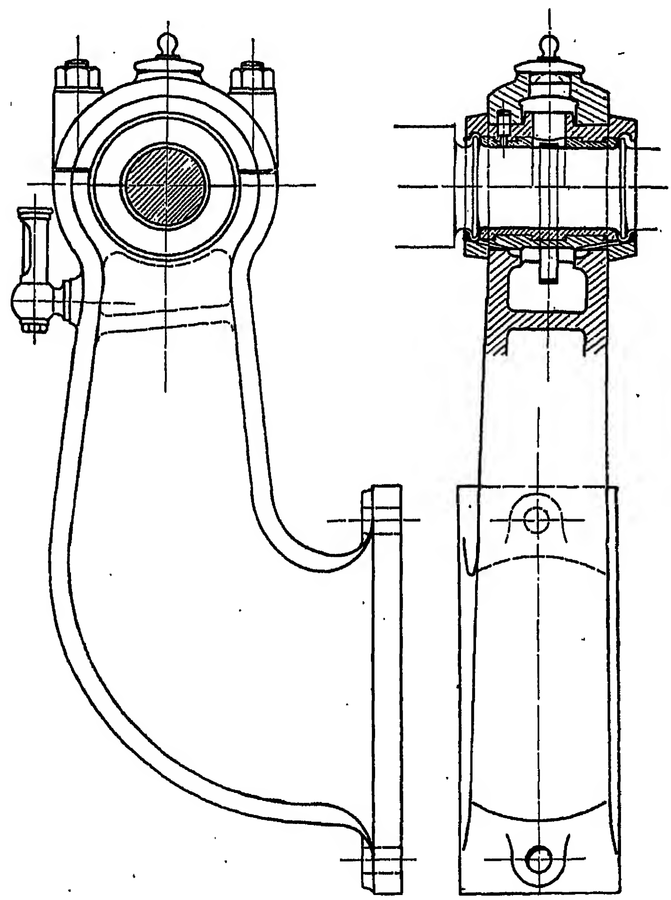


Fig. 214. Fig. 215.

Figs. 214 and 215.—Type of Horizontal Cam Shaft Bearing Brackets.

as the engine. The governor is mounted on the vertical shaft above or below this casing.

The horizontal cam-shaft bearings are often provided with ring lubrication (Figs. 214 and 215), and are carried in brackets. These brackets are bolted to the cylinders or to the framing, and may be two in number for each

cylinder, or of the same number as the cylinders plus one. Their alignment is adjusted during erection and fixed by spigots or dowel pins.

There are also examples of these brackets being secured to the cylinder head instead of to the framing, and also of the cam-shaft brackets being cast in one piece with those for the fulcrum shaft (Fig. 216). The first of these arrangements, however, is inconvenient, especially with multi-cylinder

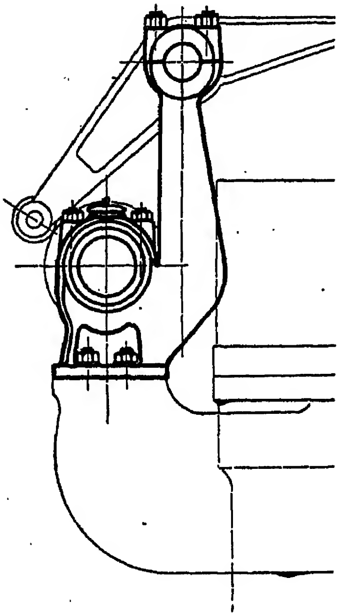


Fig. 216.—Fulcrum Shaft Bracket cast in one piece with Cam Shaft Bracket, and secured to Cylinder Jacket instead of Head.

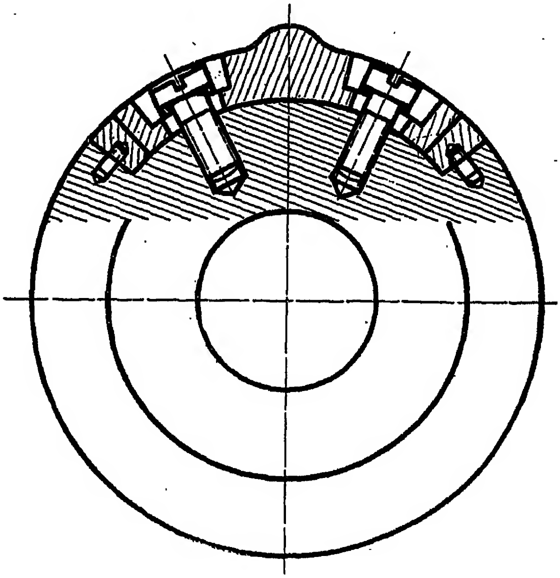


Fig. 217.

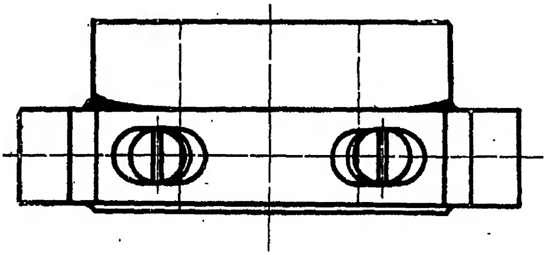


Fig. 218.—Cam for Fuel Injection Valve.

engines, since it makes it necessary to dismantle the cam shaft whenever it is required to lift off the cylinder head.

Cams.—The cams are secured to the cam shaft by keys or pins, at an angle determined as a result of the shop trials of the engine, for which trials it is sufficient to secure them simply to the shaft by set screws. The form of the suction and exhaust cams is seen in Figs. 206 and 207, p. 147, whilst that for the fuel injection valve is represented by Figs. 217 and 218. The

nose of the latter is separate from the body of the cam, and is secured thereto by two screws passing through two elongated holes in the nose piece, allowing the moment of opening of the valve to be advanced or retarded without displacing the cam on the cam shaft. From the indicator diagram, during the shop trials, the most suitable position for this nose is determined, and it is permanently fixed by two iron packing pieces, as shown in Fig. 217.

As explained when dealing with fuel injection valves, a saving in compressed air is effected at reduced loads, especially with large engines, if the lift and duration of opening of the fuel injection valve can be reduced.

Fig. 219 shows the arrangement adopted by Sulzer to serve this end. The cam of the fuel injection valve consists of two discs *e*, between which

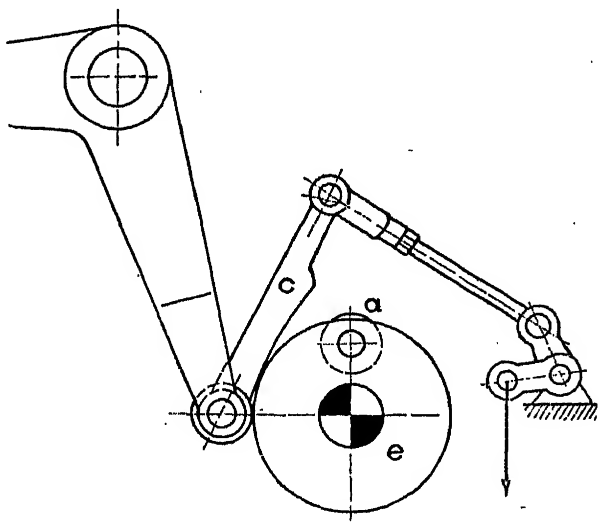


Fig. 219.—Arrangement for varying lift and duration of opening of Fuel Injection Valve (Sulzer).

the roller *a*, serving in place of the nose piece of the ordinary form of cam, is held. The fuel injection lever has two other rollers working on the discs *e*, and connected with a link *c* of special form, attached in its turn to a compressed air servo-motor acting in accordance with the load on the engine. The roller *a* passing the link *c* moves it more or less, and so gives to the fuel injection valve a greater or smaller lift according to the inclination of this link *c*.

To permit of the rapid acceleration of the engine during starting, arrangements are made in some designs to diminish the engine compression during these periods, as is frequently done with gas engines. Means are generally provided to hold open one of the valves when it is wished to turn the engine by hand to the desired crank position prior to starting, or some other method

is adopted to reduce the compression. Fig. 220 shows such an arrangement (L.W., Tosi, M.A.N.).

The cams are usually of forged steel, whilst that for starting may be of hard cast iron, as is also the fuel injection cam, which is provided, however, with a nose of hardened steel.

The profiles of the suction, exhaust and scavenging cams are almost always composed of two tangents to the base circle, meeting an arc of a circle with a radius equal to that of the base circle plus the lift of the valves. These tangents should not be too short—to avoid high velocities of lift of the valve—nor so long as to shorten too greatly the time of full opening of the valve. With normal cams the tangents make an angle of about 50°.

It is very necessary to take into account the clearance between the roller and the cam when fixing, firstly, the points at which the tangents meet the base circle and, secondly, the radius of that base circle.

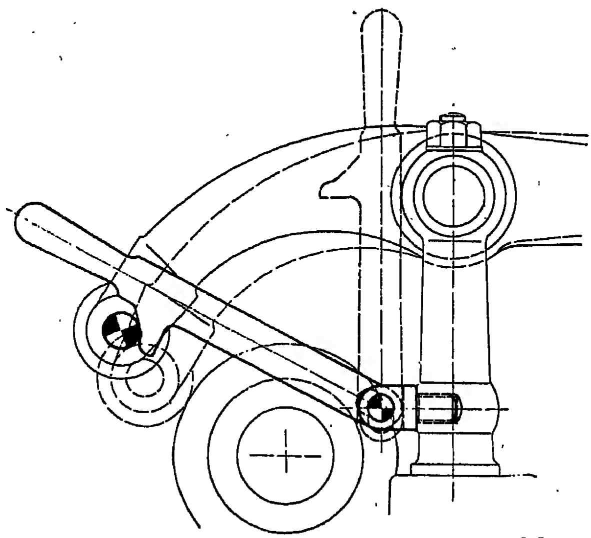


Fig. 220.—Arrangement for holding Exhaust Valve open while turning Engine by hand.

If c represents this clearance (Fig. 221), the effective angle of opening of the valve becomes d instead of e, and the value of the maximum lift is a instead of a_1. Given this method of construction of the cam profile, the lift of the valve is influenced by the value of the diameter b.

Perhaps the most rational method of determining the most suitable diameter is by comparing the diagrams for the valve lifts corresponding to the various piston positions with that of the piston velocities in these positions. The more similar these two diagrams, the more constant will be the velocity of the gas through the valve area, and the better the cam profile.

With Diesel engines, the diameter b is usually about double the smaller diameter of the valve head. Besides the type of profile described, composed

of tangents and arcs of a circle, others have been proposed; for example, the polarsinoid cam (see Fig. 222*a*).*

The necessity for clearance between the roller and the cam prevents the action of this mechanism from being gentle and silent, and indeed the profile of the nose of the cam being tangential to the base circle of diameter b, and not to a circle of $b + 2c$, causes the valve to commence its lift and to return to its seat with a finite velocity. To avoid this, the cam indicated diagrammatically in Fig. 222 † may be applied, but the necessary exactness in adjustment causes certain inconvenience, and to take account of the expansion of the valve stem under working conditions is extremely difficult, and detracts seriously from the theoretical advantages of this form.

The profile of the cam for the fuel injection valve is arrived at by means of experiments, and the opening of the valve is arranged to take place so that combustion may be produced as nearly as possible at constant pressure. Theoretically, supposing instantaneous delivery in proportion to the lift of the valve, the form of this cam should be that of a sine curve.‡

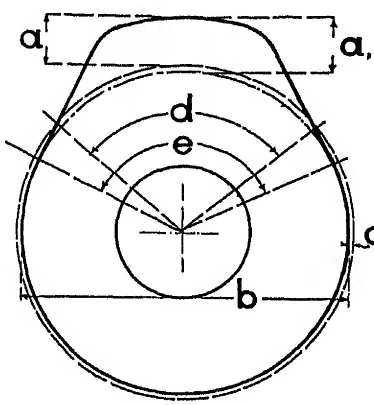


Fig. 221.

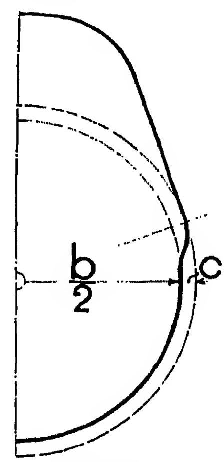


Fig. 222.

Figs. 221 and 222.—Profiles of Cams.

Having designed the cam and calculated the spring, it is always desirable, especially with high-speed engines, to make sure that the spring is strong enough to give the valve sufficient acceleration to maintain constant contact between the roller and the cam. For this to be the case, the sum of the forces acting upon the stem of the valve in the direction of opening should at any instant be less than the pressure of the spring in the opposite direction.

The forces acting upon the valve stem are the weight of the valve and the unbalanced weight of the valve actuating gear, which may be said to

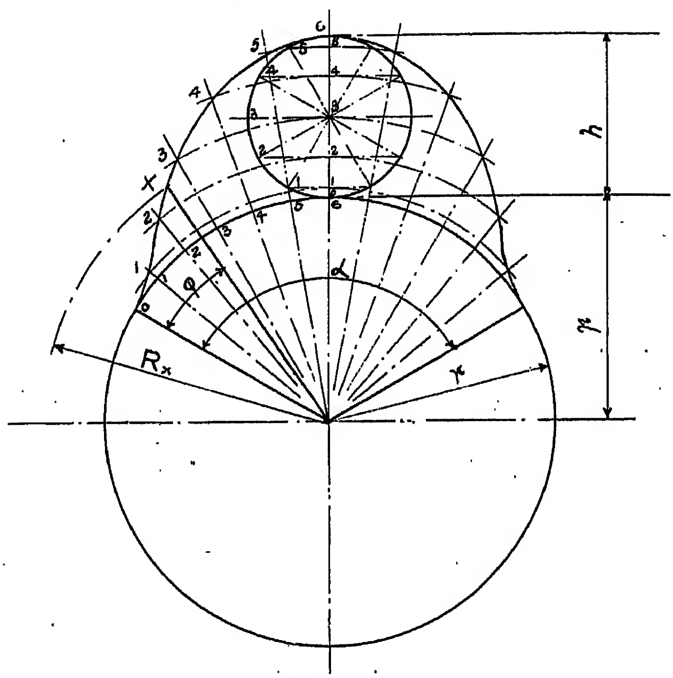
* Hartmann, *Zeitsch. des Ver. deut. Ing.*, 1905, p. 1627.
† See *Engineering*, 7th July, 1912.
‡ See Belluzzo, *Questions relating to Internal Combustion Engines*, Industria, 1912.

act on the valve centre, the friction, the difference of pressure on the two sides of the valve, and the accelerating forces of the movement given by the cam.

The calculation for the spring need not be made with extreme accuracy, and it is sufficient that the reaction of the spring should always exceed the forces stated by an amount sufficient to compensate for unforseen factors.

This question may be elucidated by diagrams, neglecting those forces of lesser importance; for example, the small loss of pressure consequent upon wire drawing of the gas through the valve opening, and also the friction.

In Fig. 223 the curve of valve movement is drawn with the abscissæ representing time and the ordinates the lifts h of the valve. From this curve



$$R_x = \left(r + \frac{h}{2}\right) - \left(\frac{h}{2}\cos\frac{2\pi}{a}\phi\right).$$

Fig. 222*a*.—Graphic Method of drawing Polarsinoid Cam.

another representing $\dfrac{dh}{dt}$ may be drawn, in which the ordinates represent the velocity v at any instant, and thirdly $\dfrac{dv}{dt}$, the curve of accelerations a *

* To construct this curve, it should be remembered that to integrate any curve whatsoever, the value of the angle between the tangent to that point on the curve and the axis of the abscissæ is taken; substituting then the arc of a circle for any part of the curve a tangent is drawn to any point p for which it is desired to find the integrated value, and this tangent makes with the abscissæ an angle α as shown in Fig. 223, the value of the tangent of which gives the new point p_1. In Fig. 223, m is the unit used for the velocity curve and n that for the acceleration curve.

may be obtained. The third curve to another scale gives the forces of acceleration ($F = C.a.m$, where C is a constant and m the mass). The algebraic summation of the ordinates of the curves of weight, friction, etc., will give a method of indicating the action of the resultant forces acting on the valve. These should be at all points less than the pressure of the spring.

The curve of the pressures of the spring is similar to that of the lifts of the valve, since the pressures exerted by the spring vary with its compression, and therefore with the lifts of the valve.

Valve Setting Diagrams.—The angle d at the centre of the cam (Fig. 221) depends upon the duration of the time of opening which it is desired to give to the valve.

The valves always open a little earlier and shut a little later than theoretically required by the cycle, in order to give sufficient time for the mass of the fluid to acquire a certain velocity, and to take advantage of the inertia effect of this assumed velocity.

Fig. 224 represents diagrammatically the movement of a point on the

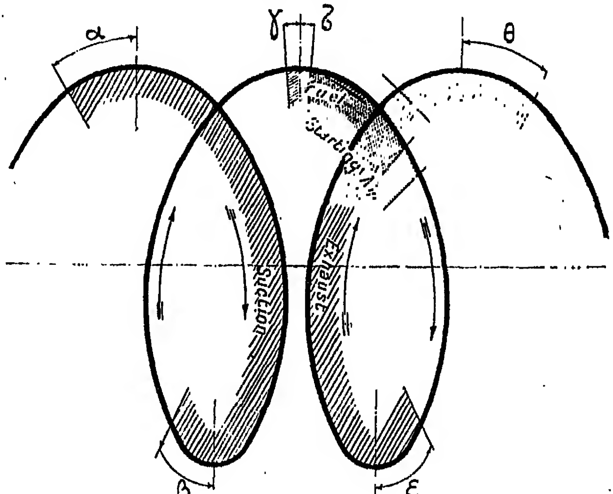


Fig. 224.—Diagram of Valve Settings referred to Crank of Four-cycle Engine.

crank circle for a four-cycle engine, supposing that a spiral replaces the circle, or, in other words, that the engine is moving axially forwards.

The angle of advance α of the suction phase is about 20°, and the angle β, during which the valve remains open after the bottom dead centre, is generally of about the same value.

The fuel injection valve opens at an angle of 2° or 3° before the top dead centre for slow-running engines, and 6° or 7° for those of high speed, and remains open for 30° to 40° after the same dead centre.

The exhaust valve opens with an angle of advance ε of 30° to 40°, and closes at an angle of θ, 10° after the top dead centre.

The starting air valve opens after the crank has passed the dead centre by a small angle δ, and with land engines remains open for about 30 per

cent. of the stroke, whilst with marine engines the time of opening depends, as will be seen, upon the number of cylinders and upon the cycle.

The same diagram of valve settings for a four-cycle engine is shown in Fig. 225, referred to the cam shaft instead of to the crank shaft. Since this cam shaft revolves with an angular velocity of half that of the crank shaft, the cycle is completed in one revolution, and the various periods of opening are given by an angle having a value equal to one-half that of the duration of the corresponding phase referred to the crank shaft, as shown in Fig. 224.

In drawing the cam, the angle d at the centre (Fig. 221) is taken. For example, for a suction valve, $d = 90° + \frac{\alpha}{2} + \frac{\beta}{2}$, and for an exhaust valve, $d = 90° + \frac{\varepsilon}{2} + \frac{\theta}{2}$.

With marine engines provision must be made so that the engine will start

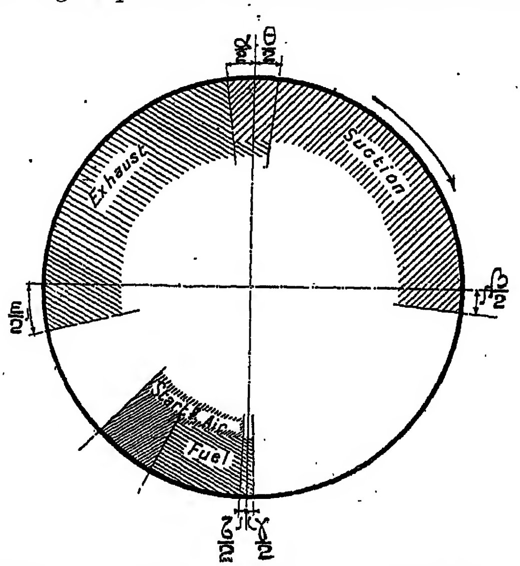


Fig. 225.—Diagram of Valve Settings referred to Cam Shaft of Four-cycle Engine.

from all crank positions, and, therefore, one cylinder must always be in the starting position with a starting air valve in action. To satisfy this condition, the angular duration of the lift of each starting air valve should be such that the sum of these angles for all the cylinders is equal or greater than the angular period of revolution of the crank shaft corresponding to one complete cycle.

In this way the starting air angle for each cylinder should be

$$\varphi > \frac{720°}{n} \text{ for four-cycle engines,}$$

and

$$\varphi > \frac{360°}{n} \text{ for two-cycle engines,}$$

in which n represents the number of cylinders.

This condition must be satisfied, but since the exhaust always commences about 40° before the dead centre, the angle available for admission and expansion of the starting air is only about 140°, and the condition that $\varphi < 140°$ must also be satisfied, to prevent the starting air valve from being open at the same time as the exhaust when compressed air would blow uselessly into the exhaust pipe.

From the foregoing it is clear that with four-cycle engines six cylinders at least are required to render starting certain, whatever be the position of the crank shaft, and then $\varphi = \frac{720}{6} = 120°$, whilst with four cylinders $\varphi = \frac{720}{4} = 180°$, which is greater than can be allowed.*

With marine four-cycle engines of less than six cylinders, starting must be carried out on the two-cycle principle, in which $\varphi =$ or $> 90°$, and for three cylinders $\varphi =$ or $> 120°$.

Since engines of less than six cylinders are always provided with a flywheel, it is sufficient that the starting on the two-cycle principle should merely give the first few impulses, and after several revolutions, when the engine has acquired sufficient speed of revolution, the starting air valves commence to open every second revolution, causing the compression strokes to take place and to raise the temperature of the air for the combustion of the fuel.

Valve Settings for Two-cycle Engines are the same whether they be referred to the crank shaft or to the cam shaft, since both turn at the same speed of revolution.

The fuel injection valve has a period of opening similar to that for four-cycle engines of the same type. The starting air valve opens in this case also some degrees after the top dead centre, and remains open for about 30 per cent. of the stroke with land engines, and, as already stated, the duration of opening is $\varphi > \frac{360}{n}$ for marine engines.

Exhaust must always take place through ports covered and uncovered by the piston, and this period represents an angle of 90° to 120° of the crank shaft rotation. The exhaust setting on the diagram is naturally symmetrical about the vertical line through the dead centres.

Scavenging may be effected by means of valves or ports, and in the latter case (Figs. 34 to 36, p. 36) the angle of duration of opening of these ports is also symmetrical about the dead centre (Fig. 226). For the reasons given on p. 36, the scavenging angle with this system must be less than that of

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* Four-cylinder four-cycle engines have been constructed (M.A.N. Augsburg, and French submarine engines) in which $\phi < \frac{720}{4}$, but in this case a clutch is necessary between the engine and the propeller. Before starting, the engine must be turned by hand if necessary, to put one of the cranks in the starting position, and the engine started with the clutch out. When the order to go ahead is given, the clutch is put into gear. Reversal must always be carried out with the clutch out of gear, unless advantage be taken when stopping, as can sometimes be done with small engines, of the half-revolution in the opposite direction caused by the engine compression. The certainty of this manœuvre depends, however, too greatly on the quickness and experience of the engine-room personnel to merit consideration.

exhaust, and in general the exhaust may be stated to commence from 10° to 15° before the scavenging ports are uncovered by the piston, and to terminate necessarily an equal period of revolution after the scavenging ports are closed.

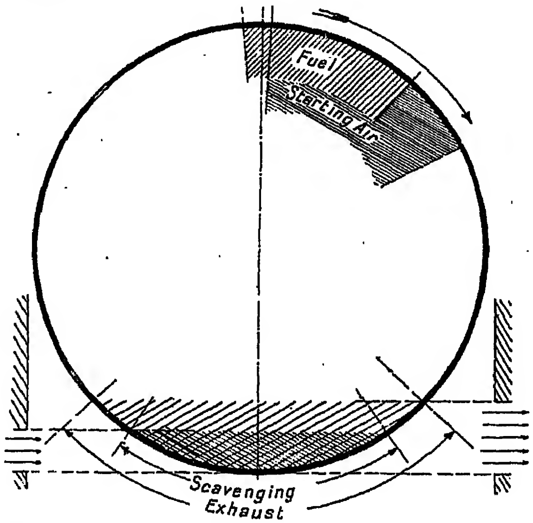


Fig. 226.—Diagram of Valve Settings of Simple Port Scavenging Two-cycle Engine.

If scavenging is carried out on the Sulzer system, with two series of ports, the upper being controlled by a valve (Figs. 37 and 38, p. 37), the scavenging air continues to enter the cylinder for 10° after the exhaust ports have been

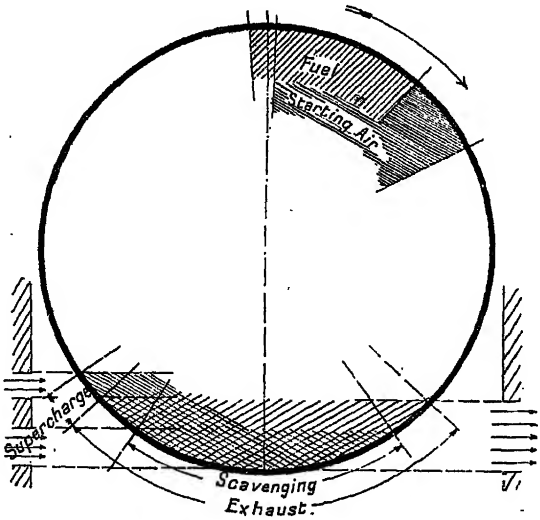


Fig. 227.—Diagram of Valve Settings of Double-port Scavenging Two-cycle Engine (Sulzer System).

closed. With a total angle of exhaust of 100°, the scavenging ports are closed by the piston 60° after the bottom dead centre. The lower or main scavenging ports commence the scavenging of the cylinder 10° after the exhaust ports have been opened, due to the valve controlling the upper ports remaining closed at the commencement of scavenging (Fig. 227).

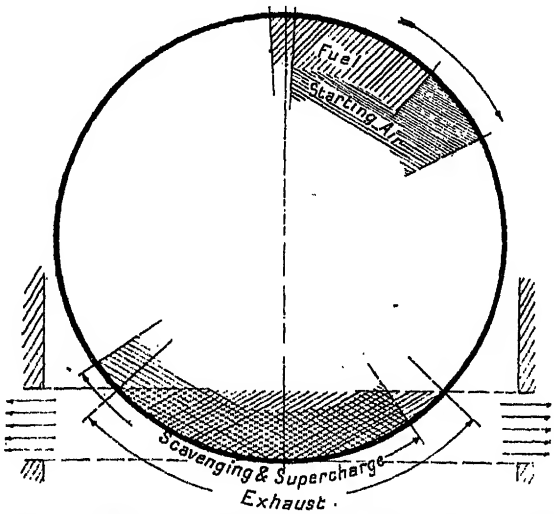


Fig. 228.—Diagram of Valve Settings of Valve-Scavenging Two-Cycle Engine.

If scavenging is carried out by valves in the cylinder head (Fig. 204), these are opened 10° to 20° after the uncovering of the exhaust ports, and are shut also 10° to 20° after the piston has covered the exhaust ports (Fig. 228).

CHAPTER XI.

THE REVERSAL OF MARINE DIESEL ENGINES.

The solutions which have been put forward to solve the problem of the reversal of motion of ships with internal combustion engines are of two kinds; first, that which permits of changing the direction of rotation of the engine, and second, that of keeping constant the direction of rotation of the engine, and reversing the direction of the propeller shaft or of the propeller itself (by means of reversible blades). Only these two solutions of the problem will be mentioned, since that of providing, as is done in the case of turbines, a separate engine for astern running may be neglected as being too costly, clumsy, and tending towards slow and uncertain manœuvring, due to the necessity of starting up a cold engine when it is desired in an emergency to go astern.

External systems of reversing—*i.e.*, those in which the direction of rotation in the engine is not changed, were the most common with early applications of the oil engine to marine propulsion. That this was so is perhaps natural, since the chief experience with internal combustion engines at that time was with petrol engines, which are almost always irreversible, and moreover, external reversal did not require any modifications of the engine itself.

With recent applications, however, internal or direct reversal is almost always preferred for reasons of lightness and compactness, and as being more rational, and, above all, because it does not impose limits to the power of the engine, whilst the greater number of the methods of indirect or external reversal do not lend themselves to use with the highest powers.

Direct reversal gives rise to the disadvantage that the engine must be stopped and put in motion in the opposite direction—an operation requiring a large consumption of highly compressed air and a certain amount of care, as well, in some cases, lacking in certainty.

Direct Reversing Gears.—The systems of internal or direct reversing are substantially different for two- and for four-cycle engines.

Figs. 229 to 232 show diagrammatically the phases of a four-cycle engine, referred to the crank shaft and to the cam shaft, for the two directions of rotation, without, however, taking into account the angles of early opening or late closing for the various valves. The cams are indicated by the arcs of circles, acting supposedly on rollers in the same plane, which is not true in practice, but does not affect the results of the following considerations.

In comparing the diagrams referred to the cam shaft, it is seen that the profiles of the groups of cams which give the two directions of rotation are symmetrical with respect to a line A-A, but cannot be superimposed.

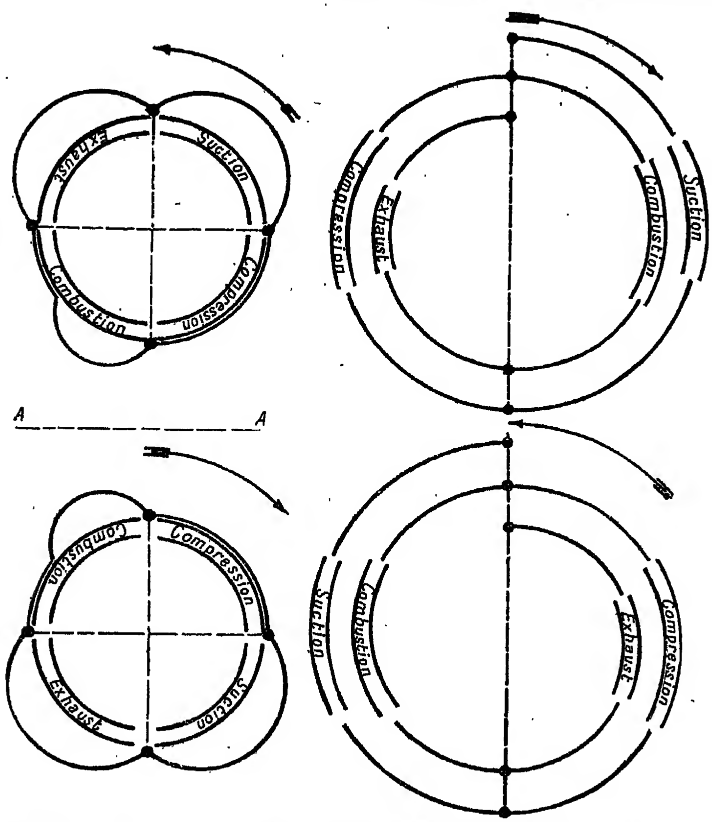
On this account four-cycle engines cannot be reversed by simply rotating

the cam shaft through an angle relatively to the crank shaft; they need two series of cams, one for ahead and the other for astern running.

Usually the two series of cams are keyed on to the same shaft, and the cams for the same valve for ahead and astern are adjacent.

To reverse the direction of rotation of the engine, all that is required is to stop the engine, move the cam shaft longitudinally, so that the rollers of the valve levers move on to the cams which correspond to the desired direction of rotation, and to start the engine again.

The fulcrum of the levers of the starting air and fuel injection valves are



Figs. 229 to 232.—Diagrams of the Phases for the two directions of rotation of a Four-cycle Reversible Engine.

worked on the usual eccentric sleeve, which permits of working alternatively with the starting air valve or the fuel injection valve in gear. These valves are put in and out of gear by one lever for all the cylinders of the engine, or by two levers each controlling half the number of cylinders. In this latter case, there is the advantage that fuel oil may be given to half the engine, whilst the other half is running on compressed air, thereby considerably facilitating starting, and the further advantage that with one of the two levers in the

intermediate position it is possible to obtain very slow running of the engine with only half the number of cylinders working on fuel oil.

As tending towards simplification, it may be arranged that when this lever is in the starting position a cock is opened controlling the compressed air from the starting reservoir to the starting air valves on the engine, whilst, when the lever is in the running position, this cock is closed, and instead the fuel injection pumps are put into action. In the intermediate position, the cock controlling the air is shut and the fuel injection pumps are out of gear.

The longitudinal movement of the cam shaft is carried out by hand in the case of small engines, and by means of a compressed air servo-motor with large engines.

It may happen (with multi-cylinder engines it always takes place) that a valve which remains shut when the engine has stopped ought to be opened for the new direction of rotation. For example, if the cylinder under consideration is on the compression stroke on stopping the engine, by the reversal of the valve gear the stroke would be changed to the suction stroke (see Figs. 229 to 232), and the suction valve which was shut when the engine stopped would be opened as a result of the reversal of the valve gear. In this way all the valve levers which are subject to these conditions—in other words, all those which for the preceding direction of rotation were shut, and for the new direction require to be opened—would seriously impede the longitudinal movement of the cam shaft, whilst the rollers of those valves which were in the position of full opening—*i.e.*, upon the noses of the cams—would fall from the edges of the cams and cause the valves to hammer upon their seats.

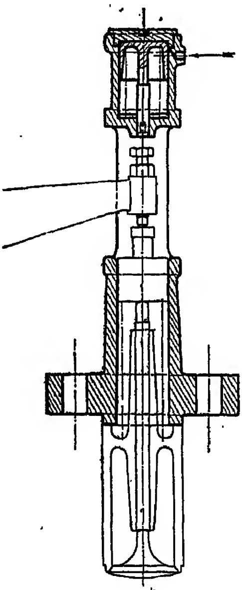


Fig. 233.—Mechanism for lifting Valve Actuating Levers during reversal (Kolomna).

For this reason, before reversal is carried out, it is necessary, prior to moving the cam shaft in a horizontal direction, to lift all the valve levers from their cams at the same moment, and to return them only when this horizontal movement is completed.

It is not difficult to devise a mechanism for accomplishing the lift of the valve levers, and many simple and suitable examples are to be found, but certain constructors prefer the more rapid gear shown in Fig. 233 (Kolomna). This consists of a small piston actuated by compressed air which opens the valve, and so lifts the valve lever and roller before the cam shaft is moved horizontally, and only allows the valve to shut and the roller to return to its cam when this manœuvre is finished.*

* If it be arranged that the longitudinal movement of the cam shaft cannot take place except the lever commanding the eccentric sleeve of the starting air and fuel injection valves be in the intermediate position, it is not necessary to lift the levers of these valves, since their rollers are already clear of the circle described by the points of the cams.

Instead of moving the cam shaft longitudinally, in order that the valve levers may be acted upon by one or other of the series of cams, recourse may be had to other mechanical arrangements.

Fig. 234 shows a system applied to some Nobel engines. Each valve lever is actuated through a link connected to a triangular piece carrying rollers c_1 and c_2 vertically disposed over the cams A V and A D, but with their axes out of line with one another. By rotating the manœuvring shaft m, the roller c_1 is brought into contact with the cam A V, or c_2 into contact with A D. Only one roller can be in contact with its cam at any time.

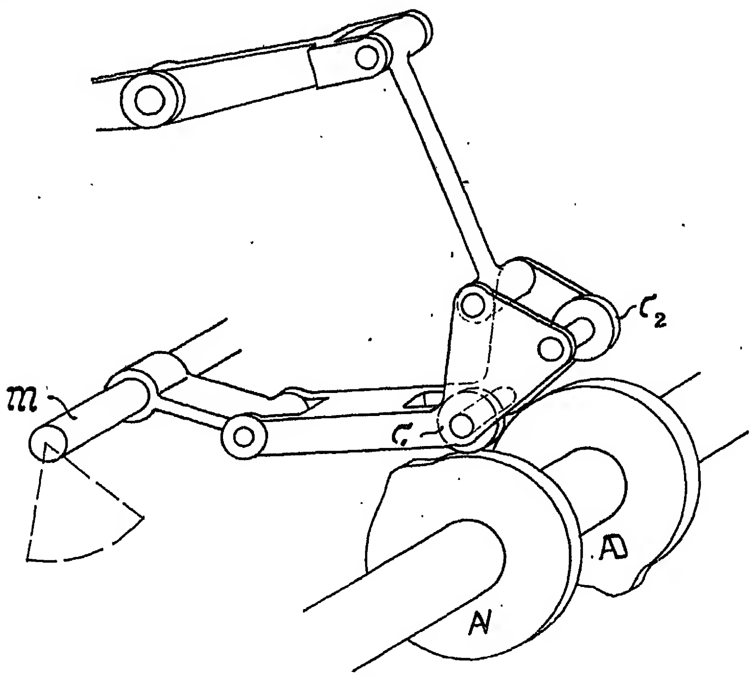
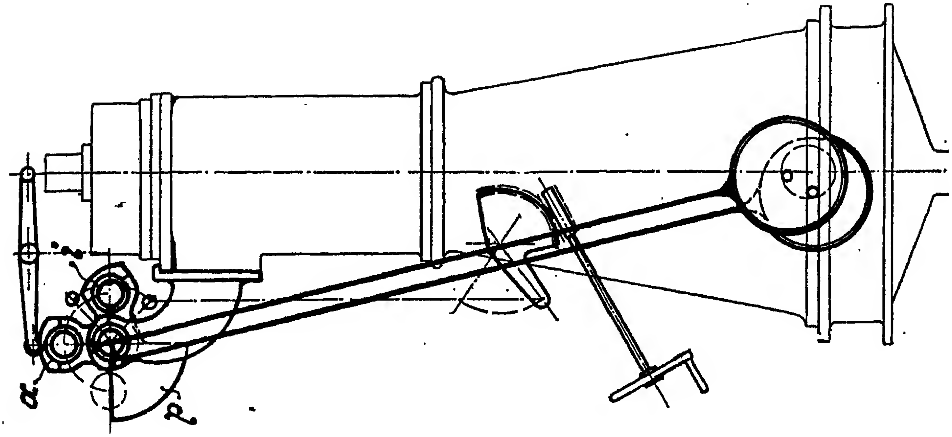


Fig. 234.—Nobel's arrangement for reversing Four-cycle Engines without reversing the Cam Shaft.

The system applied to some M.A.N. Augsburg, engines is shown in Fig. 235.

In this type also two cams are each in line with one or other of the two intermediate rollers, one of which transmits the lift of its cam to the valve lever roller, whilst the other remains free. The roller of the lever is of such a width as to enable it to take its motion from either of the intermediate rollers. A manœuvring lever, having its fulcrum on the cam shaft, permits of changing the intermediate roller, and so the cam for operating the valve, and in this way the direction of rotation of the engine is reversed.



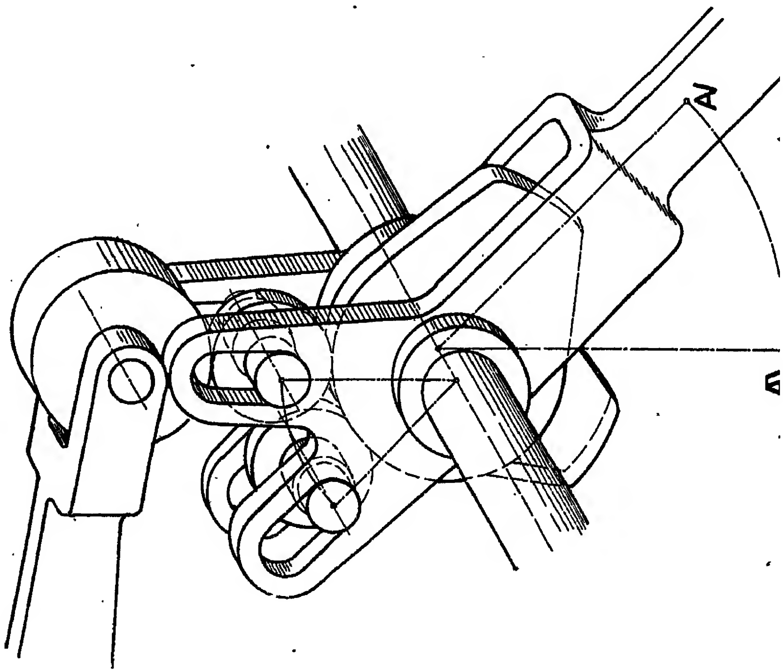


Fig. 236 represents the general arrangement of the reversing gear of the Werkspoor engine of the motor ship "Vulcanus." There are two cam shafts *a* for ahead and *i* for astern running. The supports for these shafts are connected by angled brackets, which have their fulcrum on the main horizontal shaft, around which they can rotate, so that the cam on the shaft *a* or on the shaft *i* acts upon the valve lever according to the direction of rotation required.

Counter weights *p* balance these angled pieces and the two cam shafts, which latter receive their motion from the main horizontal shaft by means of spur wheels, reducing the speed of revolution to one-half. The main horizontal shaft can be driven from the crank shaft in the usual way by means of spiral wheels, or, as is done in this case, by means of eccentrics and rods,

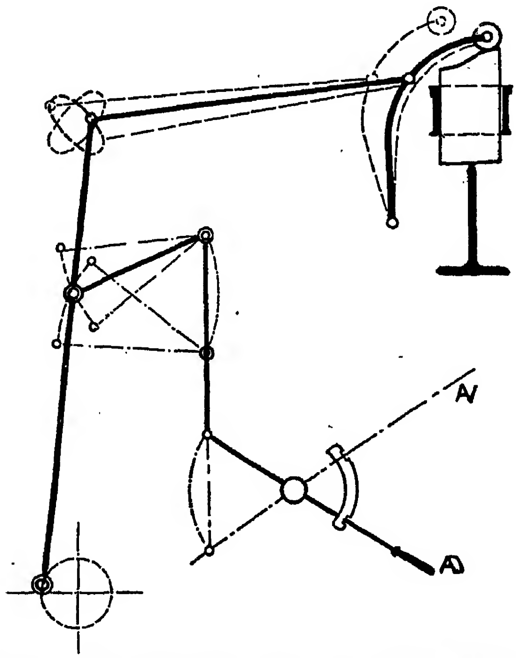


Fig. 237.—Diagram of Reversible Valve Gear derived from Marshall's Valve Gear.

giving it the same speed of rotation as the crank shaft. This system, although sufficiently reliable, cannot be said to be either light or economical.

In the latest types of construction by this firm, two parallel cam shafts are also provided, but the substitution of one or the other underneath the valve levers is a straight line movement instead of rotation about a fixed centre.

The adoption of a double series of cams for accomplishing reversal of four-cycle engines, although not the only possible method, is that which finds by far the most frequent use.

By making use of shaped surfaces of the type found in the Lentz valve gear instead of cams, and by controlling the movement of these parts by one of the various reversing systems used with steam engines, the reversal of internal combustion engines can also be effected. The diagram (Fig. 237)

shows an example of this type based on the Marshall valve gear, and experimented with by the firm M.A.N. Augsburg.

A similar application, it is believed, has also been tried by the Gasmotorenfabrik Deutz.

Yet another device to carry out reversal of Diesel engines consists in that shown in Fig. 238, which was applied to Thornycroft paraffin engines and to M.A.N. Augsburg, Diesel engines. With this mechanism the cam shaft always rotated in the same direction whatever might be that of the crank shaft, or, in other words, the direction of rotation of the cam shaft was changed with respect to that of the crank shaft. This system has been abandoned, however, as it does not give sufficient reliability in manœuvring.

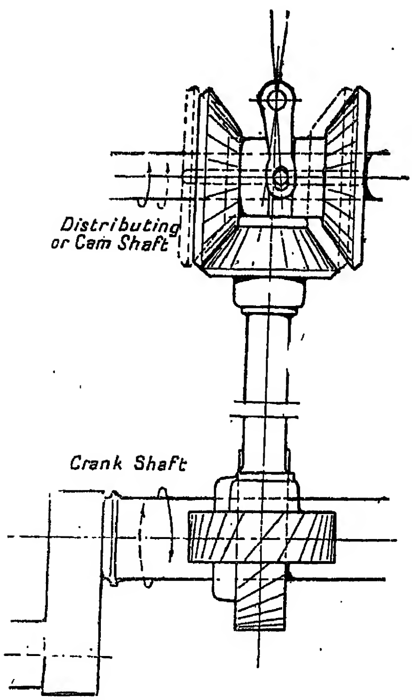


Fig. 238.—Discarded system for reversing Valve Gear by means of two Sliding Bevel Wheels.

Reversal of Two-cycle Engines.—The diagrams of valve settings for ahead and astern running of two-cyle engines are shown in Figs. 239 and 240, and, as will be seen, are symmetrical.

The exhaust, which takes place through the cylinder exhaust ports, cannot and does not require to be reversed; neither does the scavenging, if it is carried out by ports without supercharge after the closing of the exhaust ports. The valve gear of the fuel injection, starting and scavenging valves, if the latter are in the cylinder head, must, on the other hand, be reversed, as well as those for suction and delivery of the scavenging pump, unless they be of the automatic type.

Since the profiles of all the cams lie about an axis of symmetry, which coincides with the bisector of the angle of the period of opening the valves, it is clearly seen in Figs. 239 and 240 that reversal can be effected by an angular displacement of the cams with respect to the crank shaft. By rotating the fuel injection cams by an angle α, the scavenging cams by an angle β, and those for starting by ε, the direction of rotation of the engine is reversed.

If the duration of the various phases of the cycle can be so fixed that the angles α, β, and ε are equal, reversal can be obtained by a simple rotation of the cam shaft with respect to the crank shaft through an angle of $\alpha°$.

The duration of the phases and their angles depend, however,

upon the type of the engine, and more especially upon its speed of rotation, and, therefore, this condition of equal angles can only be

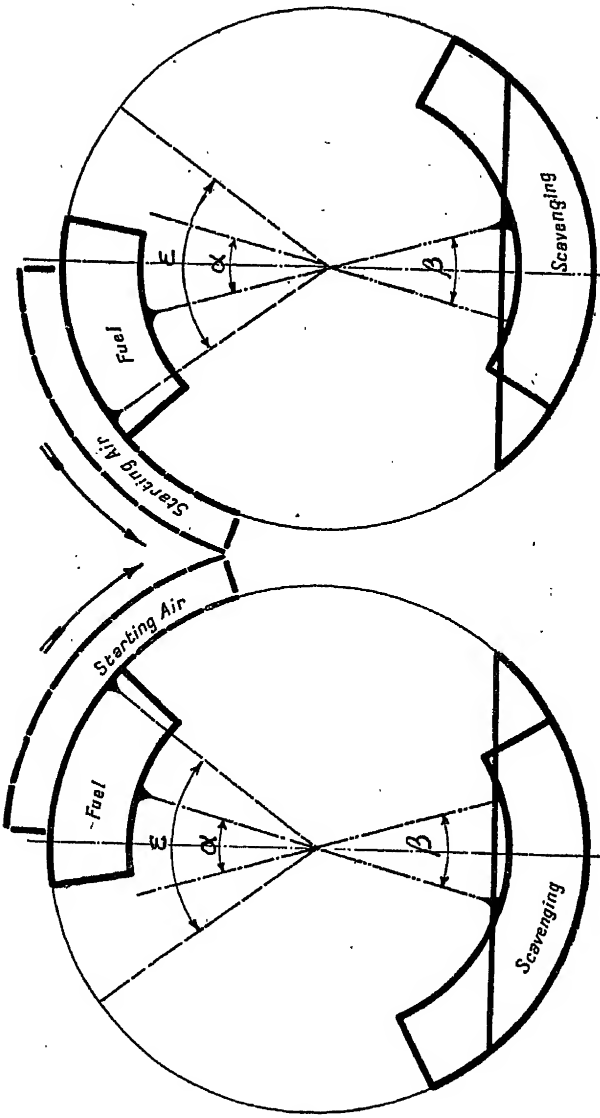


Fig. 240.

Fig. 239.

Diagram of the Valve Settings for Ahead and Astern Running of a Reversible Two-cycle Engine.

obtained in very particular cases, and never without it being necessary to some extent to modify the angle which would be most suitable

for one or other of the phases, more particularly for that of the starting air valve.*

The relation $\alpha = \beta$ between the angular displacement of the scavenging and fuel injection cams can, on the other hand, often be accomplished, and with high-speed engines it causes no inconvenience.

Since the same angle may also be assigned to the scavenging air pump, suction and delivery valves, with high-speed and light engines, the process of reversal is generally reduced to that of giving the cam shaft an angular displacement α with respect to the crank shaft, and of substituting a second set of starting air cams (Fiat, Tosi, etc.), by means of one of the methods described for four-cycle engines.

The new starting air cam, however, after the angular displacement of the cam shaft through β° would not be changed by ε° with respect to its position for the preceding direction of rotation, but only by $\varepsilon^\circ - \beta^\circ$, and since this difference is always small, instead of substituting a second set of cams,

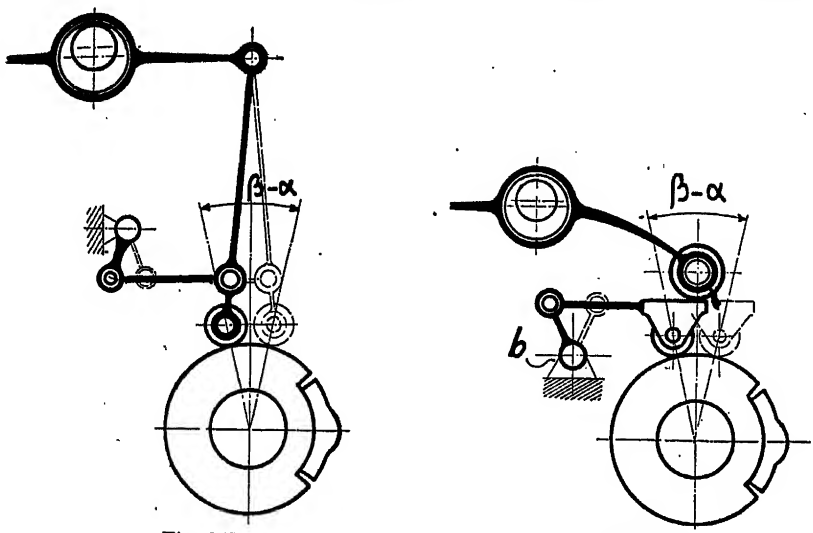


Fig. 241. Fig. 242.

Figs. 241 and 242.—Systems for correcting the angular displacement of Cam Shaft to suit Starting Air Valve or Fuel Injection Valve.

correction can be made to the valve roller by a mechanism of the type indicated in the diagrams (Fig. 241 and 242).

For slow-running engines, if the value of α° equals that of β°, the fuel

* Complete reversal by changing only the direction of rotation of the cam shaft is not applied, so far as is known, except to Körting submarine engines, in which $\alpha = \beta = \varepsilon = 43^\circ$.

However, in applying this system, it is necessary to assign to the starting air valve a very considerable angle of lead, and since this angle of lead on starting would cause the manœuvre to fail, the cam shaft must first be rotated through an angle greater than α, sufficient to ensure the opening of the starting air valve at the top dead centre. After several revolutions, when the engine has attained a certain speed, the cam shaft is brought back to its normal running position.

injection valve would generally be found to have too great an angle of lead and on that account it is often desirable to correct the position of the fue injection roller by an angle $\beta - \alpha$ after the cam shaft has been rotated througl the desired $\beta°$. This operation is similar to that described for starting ai valves (Carels, Sulzer).*

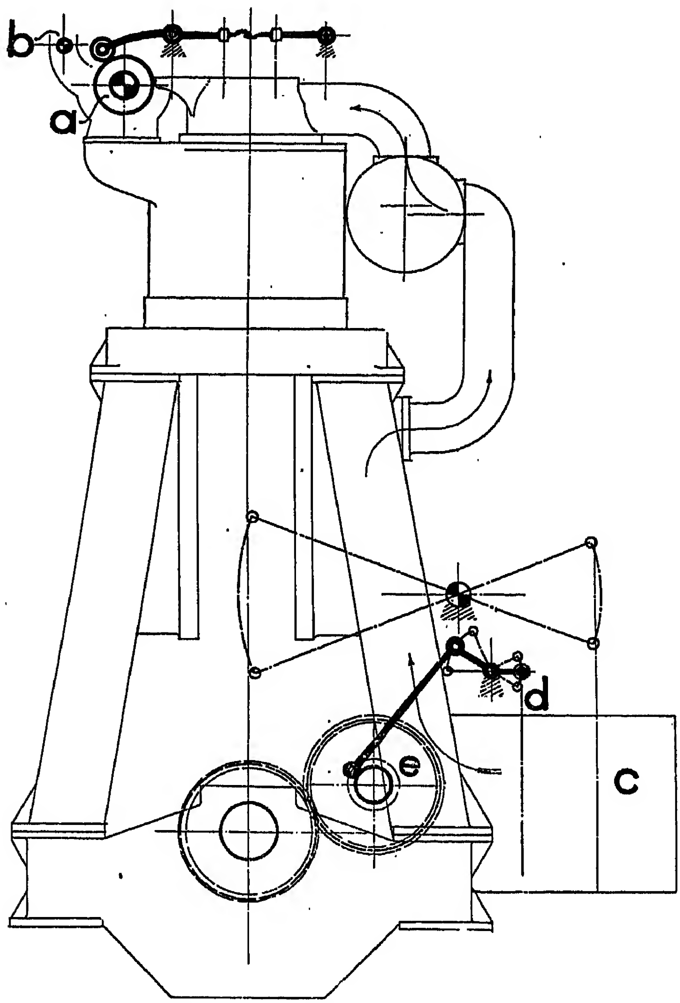


Fig. 243.—Diagram of Carels' Reversing Gear.

The foregoing forms the general criterion upon which all systems of reversal are based, and in the following part of this chapter some examples

* By this same gear the angle of lead of the fuel injection valve can be varied to correspond to the speed of revolution and the power desired from the engine.

will be given of the designs of reversing gears of the principal types of Diesel engines.

Carels Frères, Ghent.—Engines of the motor ship "Excelsior," constructed by the Reichestieg Schiffswerfte, of Hamburg, under licence from Messrs. Carels; 1,800 to 2,000 B.H.P., six cylinders, 90 to 100 revs. per minute. The scavenging is carried out by four valves symmetrically placed in the cylinder head (Fig. 204, p. 145).

Reversal is accomplished in the following way:—Upon the vertical shaft which drives the cam shaft *a* (Fig. 243) is keyed a long sliding helical wheel movable axially by a fixed amount (Fig. 244).

The displacement of this spiral wheel is carried out by a compressed air servo-motor, and gives to the cam shaft *a* a rotation of $\beta°$, required for the reversal of the four scavenging valves of each of the cylinders.

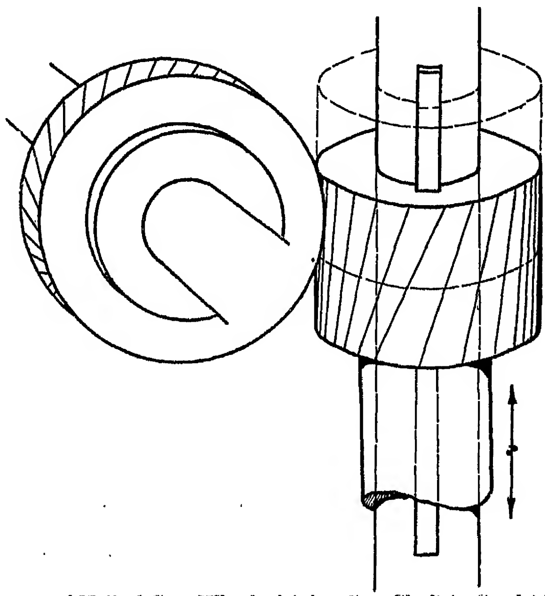


Fig. 244.—Diagram of Helical Gear Wheels driving Cam Shaft in Carels' Reversing Gear.

The correction $\beta° - \alpha°$ of the fuel injection cams, and $\beta° - \varepsilon°$ of the starting air cams, is carried out by a similar mechanism to that shown in Fig. 242.

The manœuvring shaft *b*, controlled by a hand-wheel, gives this desired correction, and its small cranks are so disposed that the following sequence is obtained:—

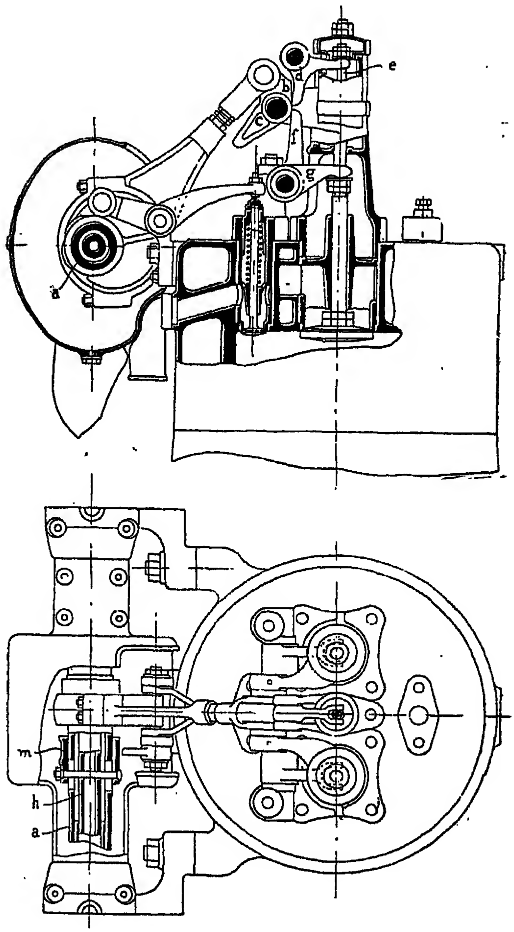
(1) Stop.
(2) Six cylinders on compressed air.
(3) Cylinders Nos. 2, 4, and 6 on air, and the remaining three on fuel.
(4) Nos. 1, 3, and 5 on air, and the remaining three on fuel.
(5) Six cylinders on fuel.
(6) Three cylinders on fuel and the other three running idle.

The cylinders Nos. 2 and 3, at the forward end, drive the double-acting scavenging pumps *c* by means of balanced levers (Fig. 243, p. 171). These pumps are provided with piston valves for suction and delivery, the piston-rods *d* of which are actuated by two cranks, driven by a spur-wheel *e* in gear with a similar wheel on the crank shaft. The wheel *e* can drive in both directions of rotation, but instead of being rigidly keyed to its shaft, it is free for a certain angle necessary for reversal of the scavenging pump piston valves. When the engine is started in a contrary direction to that in which it previously ran, the valves of the pump *c* remain inoperative for a fraction of a revolution, and only when the crank shaft has rotated through the stated angle does the spur-wheel commence to drive the piston valves.

In July, 1912, the author saw this engine running on the test bench, and reversal was accomplished with absolute certainty and rapidity.

Fiat, Turin.—Figs. 245, 246, and 247 represent the valve gear of a Fiat engine.

Since in the valve setting diagram $\alpha = \beta$, and the bisectors of the angle of opening of the fuel injection valve and of the two scavenging valves are at 180°, one eccentric with intermediate rolling levers *b*, *c*, *d*, and *f* drives all these valves.



Figs. 245 and 246.—Early Type of Fiat Valve Gear.

By causing the cam shaft to rotate through an angle β with respect to the crank shaft, reversal of the fuel injection and of the scavenging valves is effected without other provision.

The starting air valves can be operated by two cams, one for each direction of rotation, separated by a cylindrical part on which the roller runs idly when

the engine is in its normal running condition. To reverse, it is only necessary to displace the starting air cams on the hollow cam shaft a (Fig. 246) by means of the shaft b which traverses it internally. During the operation of exchanging the starting air cams, it is not necessary to provide a gear (as in the case with four-cycle engines) to lift the levers of these valves, since the lift of the starting air valves is always so small that it is only necessary for the two profiles to be sloped down to the intermediate cylindrical part to permit of the roller easily mounting its cam.

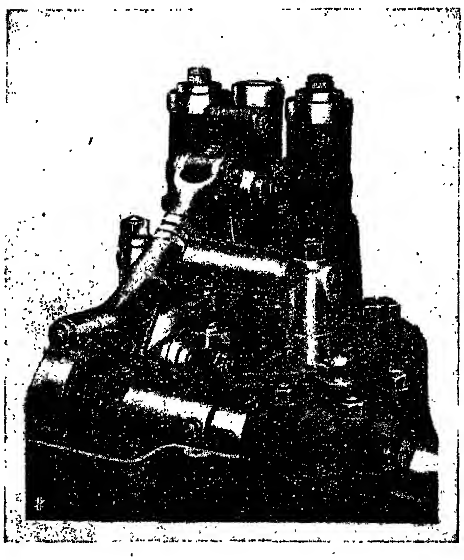


Fig. 247.—Early Type of Fiat Valve Gear.

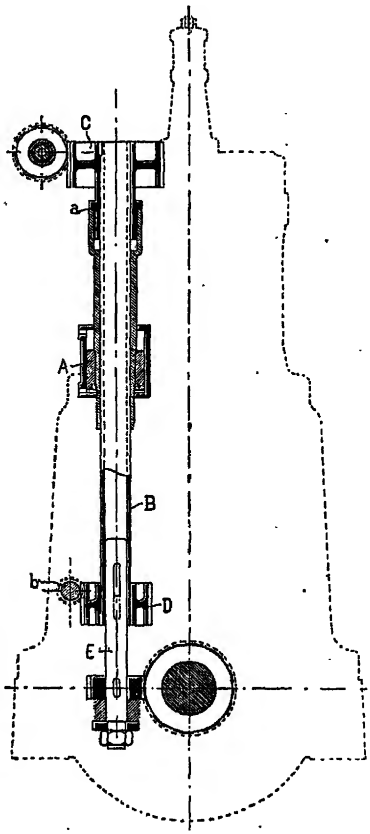


Fig. 248.—Diagram of Intermediate Vertical Shaft, with Servo-motor for displacing axially the Helical Wheels C and D when reversing Engine (Fiat).

Rotation of the cam shaft through $\beta°$ is obtained in this case also by an axial displacement of the spiral-wheel c on the vertical shaft (Fig. 248).

This same motion serves, by means of the helical-toothed wheel *d*, to effect the reversal of the shaft *b* driving the piston valves of all the scavenging pumps. For engines of high power the contemporaneous displacement of the two spiral-wheels is effected by the compressed air servo-motor A, and the complete manœuvre of reversal is carried out by a single controlling hand-wheel in the very short time of five seconds.

Prof. Junkers, Achen.—The action of the engine invented by Prof. Junkers is to be seen in the diagrams (Figs. 39 to 44, pp. 38 and 39) that have been described on pp. 39, *et seq.* Several constructors have adopted this system (Gebr. Klein, A. G. Weser, I. Frerichs, A. E. G. Nobel, Badenia, etc.).

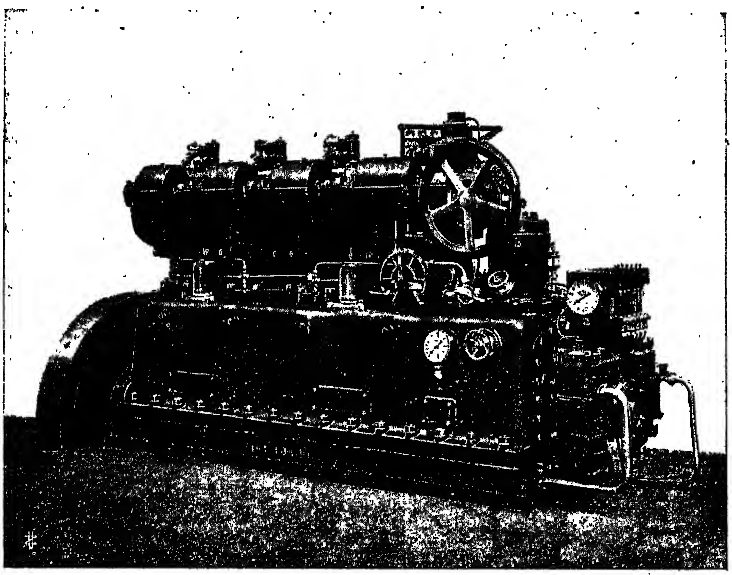


Fig. 249.—Fiat Reversible Marine Diesel Engine.

In Junkers' engines the scavenging air ports are closed by one of the two pistons of the engine before those of the exhaust, so that there is no supercharge and the scavenging does not require to be reversed. In addition, the valve gear of the scavenging air pumps does not require to be reversed, since the valves are of the automatic type.

To effect reversal, it is sufficient then to change the angles of the starting air and the fuel injection valves (of which latter there are generally two per cylinder). A system of substitution of cam shafts, similar to that described for Werkspoor four-cycle engines is adopted, and Figs. 250 to 254 show clearly the valve gear.

Gebr. Körting, Hanover.—In Körting engines (Figs. 255 and 256) the underlying principle of the valve gear is, as already stated in the note on

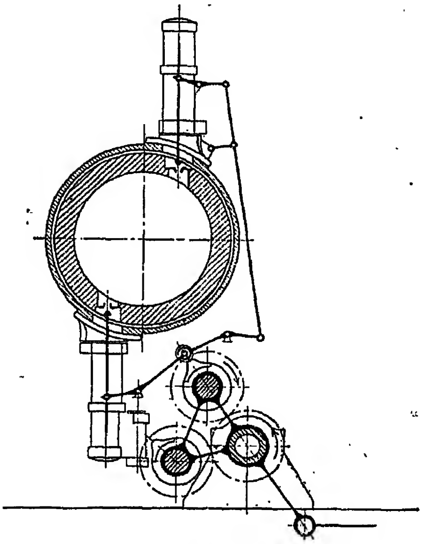


Fig. 250.

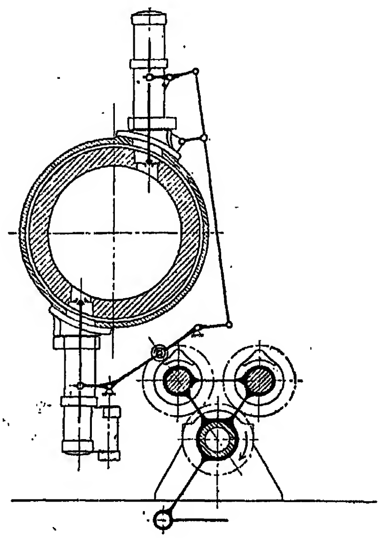


Fig. 251.

Diagrams of Ahead and Astern positions of Junkers' Fuel Injection Valve Gear.

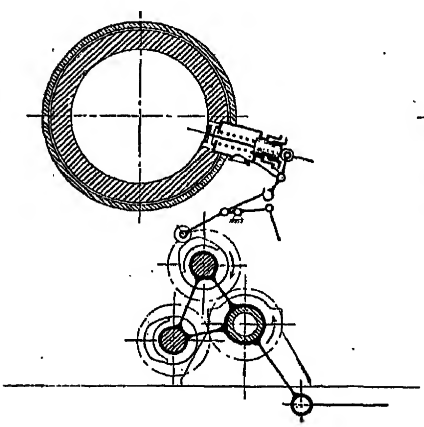


Fig. 252.

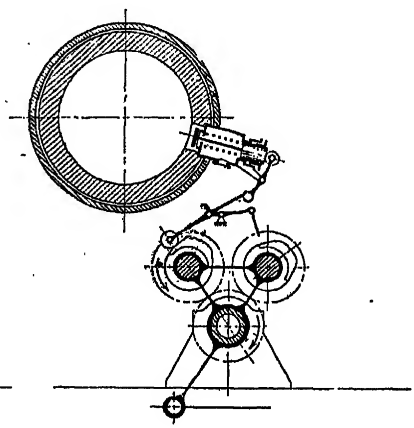


Fig. 253.

Diagrams of Ahead and Astern positions of Junkers' Air-starting Valve Gear.

p. 170, such that $\alpha = \beta = \varepsilon$—*i.e.*, complete reversal of rotation is obtained by a simple angular displacement of the cam shaft.

By turning the hand-wheel D (Fig. 256) * the leading screw E effects the axial displacement of the cross-shaft C, which, by means of the usual long spiral-wheel, produces the required rotation of the cam shaft B.

At the same time, by means of another leading screw F, the rotation of the hand-wheel carries out a similar displacement in the sliding shaft G, which changes the angle of lead of the shaft H driving the piston valves of the scavenging pumps.

As is seen, the principle on which this system of reversal is based recalls that adopted by the Fiat, although, perhaps, more attractive and expedient.

The lever L (Fig. 255) controls the usual eccentric fulcrums of the fuel

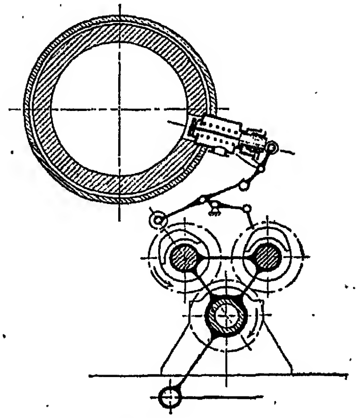
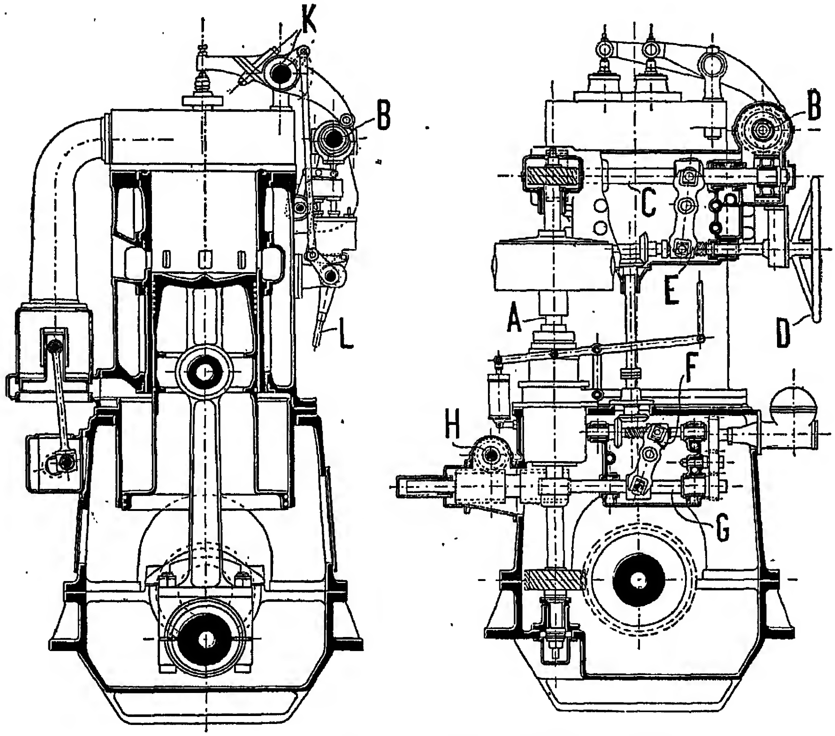


Fig. 254.—Diagram of Running Position of Junkers Air-starting Valve Gear.



Figs. 255 and 256.—Reversing Gear of Körting Marine Diesel Engine.

* *Schiffbau*, 24th May, 1911.

injection and starting air valve levers, permitting of the cutting-in and cutting-out of one or other.

M.A.N. Nürnberg.—Of the two works at Augsburg and Nürnberg, the first specialises in the construction of four-cycle engines, and the second in that of those working on the two-stroke principle. For light Nürnberg engines, the cam shaft runs centrally along the cylinder heads, and by means of two short levers controls the fuel injection and scavenging valves, one of each type to each cylinder (Plates XVIII. and XIX., facing p. 286).

The starting air valves are controlled by compressed air in quite an original way, differing entirely from those so far described.

The scavenging pumps are provided with automatic valves which do not require reversing, and, since for the fuel injection and scavenging valves $\alpha = \beta = 30°$, reversal is accomplished by rotating the cam shaft through this angle.

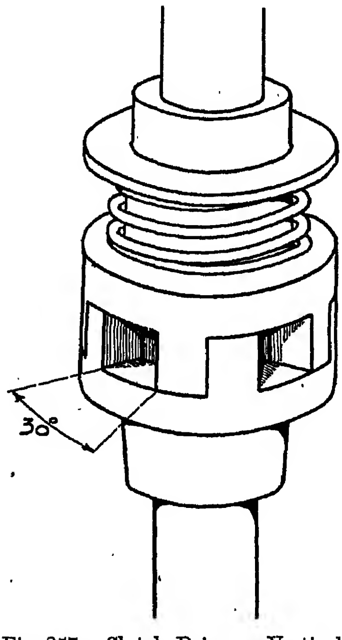


Fig. 257.—Clutch Drive on Vertical Shaft giving Angular Displacement of Cam Shaft when reversing (M.A.N.).

The vertical shaft driving the cam shaft is in two pieces connected by a dog clutch, not designed to secure rigid connection, but allowing a play of 30° between the teeth (Fig. 257). When the engine is started by means of the starting air valves in a direction of rotation opposite to that of the preceding run, the upper part of the vertical shaft, and with it the cam shaft, are not moved by the crank shaft until the latter has revolved through 30°, and taken up the play between the teeth of the dog clutch.

When the fuel injection pumps are put into gear, the engine is set without other regulation for the required direction of rotation, since the valve gear has automatically given the angular displacement necessary.

To avoid the possibility of chattering between the teeth or irregularities in transmitting the drive through the clutch, a strong spring presses the upper and lower parts of the clutch against one another, resisting in this way the inertia and accelerating forces which might tend to separate the driving faces of the teeth.

The control and the system of reversal of the starting air valves are pneumatic, and are shown diagrammatically in Fig. 258 in the position of starting for ahead running.

The manœuvring lever is put hard over to the end of its sector, and presses on the stems of the two control piston valves M and T. The cylinder T contains a valve with a conical seat, which, when it is opened, allows compressed air from the lower side of the piston of the balanced valve A to escape, and causes it to open under the influence of pressure from the starting air reservoir. Then air from the starting air reservoirs fills the

piping up to the cylinder starting air valve, but this is not sufficient to open it.

At the same time the manœuvring lever presses on the stem of the control

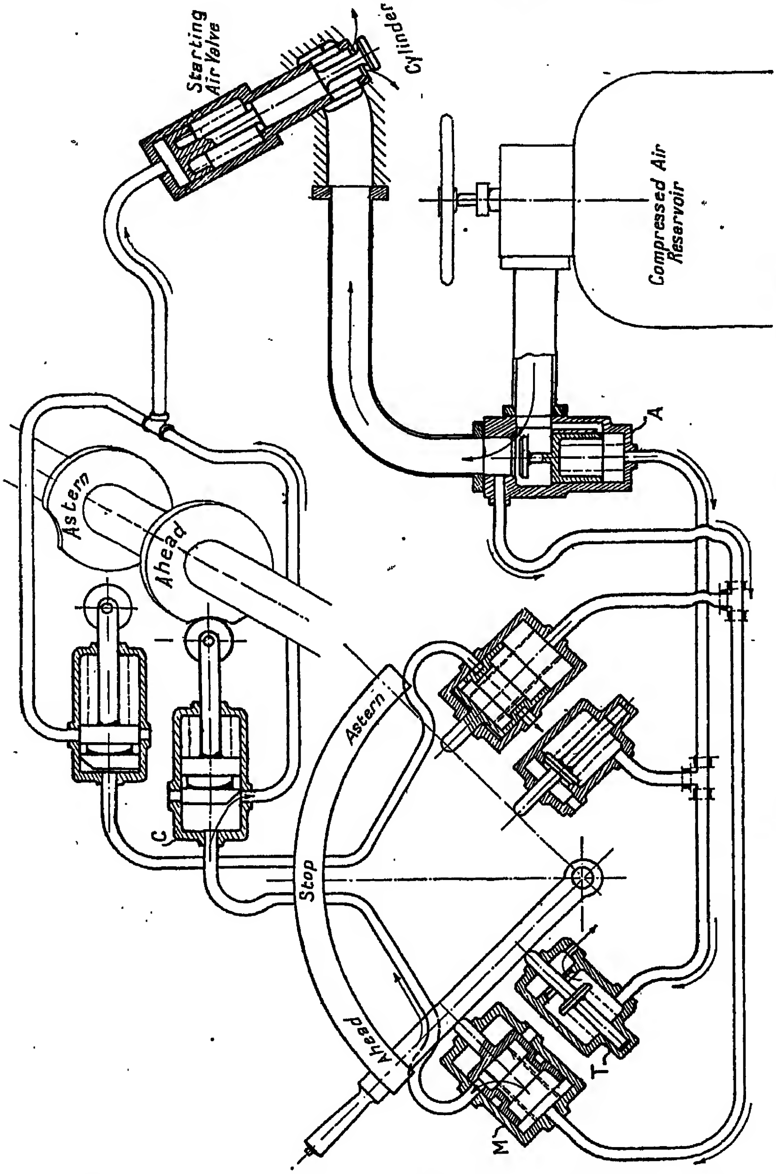


Fig. 258.—Diagram of Control Mechanism and Reversing System of Starting Air Valves (M.A.N. two-cycle engine).

piston valve contained in the cylinder M, and it is displaced so as to permit communication between the starting air lead, already under pressure, and the cylinder C.

The air reaches the piston valve, pushes it outwards, and in this way carries the roller on its plunger into contact with the respective cam. Two starting air cams are provided for each cylinder, one for each direction of rotation. The cylinder or cylinders which are in a starting position have the ahead cam in such a position that, if the roller rests upon it, compressed air through the piston valve may reach the small piston of the starting air valve and open it (Fig. 258).

Then the air from the reservoirs which has passed through A is ready behind the valve head, and enters the cylinder of the engine, and the engine runs on starting air as long as the manœuvring lever is kept at the ahead extremity of its sector, the distributing piston valve C opening the starting air valve whenever the cam presents the cut-out portion to the roller. When the lever is lifted so that it frees the stems of the small piston valves M and T, the springs close these valves.

The control valve M closes the communication between the starting air piping and the cylinder C, opening instead communication between C and the atmosphere, and the rollers are drawn back from their cams by the springs. The valve T closes the exhaust of air from the lower part of the piston of the valve A, and this also closes under the action of its spring, the pressures above and below the piston being again in equilibrium.

The manœuvring lever is also connected to the fuel injection pumps in such a way that when the starting air gear ceases to operate, the fuel oil commences to be injected into the cylinder and normal running begins. The delivery of these pumps is a maximum when the lever is near the extreme position just clear of the starting air control valve spindle, and is zero when the lever is vertical.

With M.A.N. engines, the reversal, stopping, starting in either direction, and the various speeds of revolution, are controlled by a single manœuvring lever.

Experiments made in 1911 * on a 800 B.H.P. engine showed that the engine could be put from stop to full power in six seconds, and reverse from full ahead to full astern in twelve seconds.

The M.A.N. have already experimented with two-cycle double-acting engines for large powers, regarding which nothing has so far (1915) been published.

Polar Engines, Stockholm.—The Polar engines of the Aktiebolaget Diesels Motorer of Stockholm have a system of reversal entirely different from those described, and known as the Hesselmann system.† The scavenging air enters the cylinders by means of ports, covered and uncovered by the piston (Figs. 34, 36, p. 36), and hence the scavenging setting is irreversible. Starting is not effected by allowing compressed air to act in the main engine cylinders, but is carried out by a special apparatus known as a manœuvring engine, which, during the running of the engine, serves as a scavenging pump.

Since there are no starting air nor scavenging valves in the cylinder head, only the setting of the fuel injection valve has to be changed for the direction of rotation, and this is easily accomplished by a substitution of cams.

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* Kaemmerer, *Die Verwendung von Dieselmaschinen zum Antrieb von grösseren Seeschiffen.*

† The Benz Company, of Mannheim, have taken out a licence to construct marine engines of this type.

As regards the manœuvring engine, this consists of two double-acting cylinders *c c* (Fig. 259) with cranks at 90° and piston valves. During the normal running of the engine, the cylinders *c c* serve as scavenging pumps,

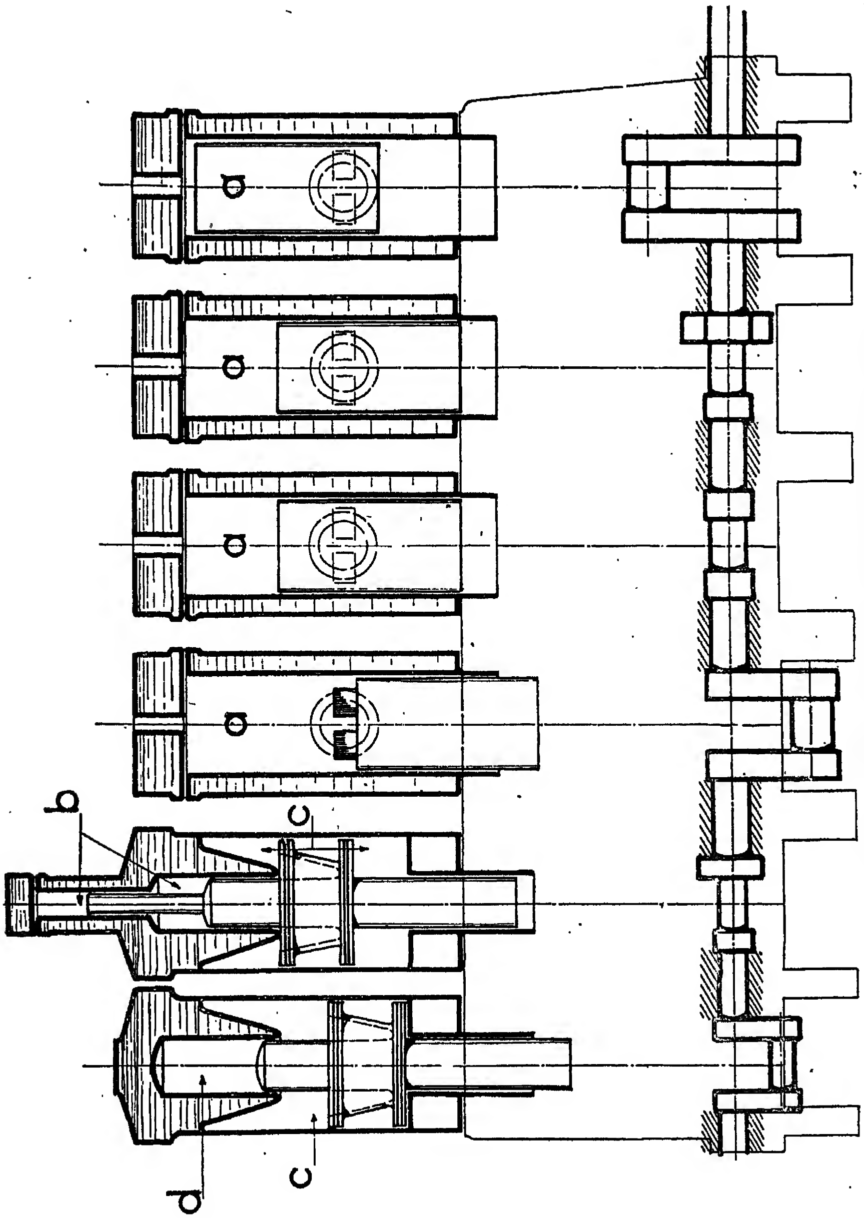


Fig. 259.—Hesselmann System of Reversing on Polar Engine.

and compress the air to 0·15 to 0·20 atmosphere (2·1 to 2·85 lbs. per square inch), whilst during the starting period the same cylinders work as compressed air engines deriving their source of power from the compressed air stored in reservoirs, delivered through the piston valves after passing through a reducing valve to lower the pressure to about 5 atmospheres (70 lbs. per square inch). When the engine has started, the compressor *d* recharges the starting reservoirs, and automatically ceases compression when the desired pressure has been reached in these reservoirs. The two-stage compressor *b* supplies the fuel injection air for the main engine cylinders *a a a a*.

Reversal of the manœuvring engine does not present any great difficulty, and is of the type common to steam engines. As has been seen, the valve gear of the main cylinders *a* is reduced to that necessary for the fuel injection valves, and reversal is very simply effected.

The possibility, peculiar to this system, of commencing the introduction of the fuel oil at the same time as the compressed air is in action, gives the considerable advantage of quick manœuvring capacity.

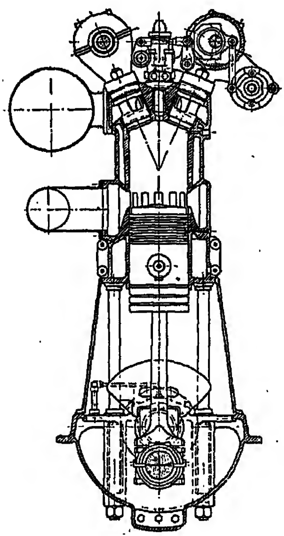


Fig. 260.—Early Type of Sulzer High-speed Marine Diesel Engine.

Since, however, the fuel injection valve lifts even when the main engine is turning under the action of the manœuvring engine, certain precautions are necessary to prevent premature ignition. For this purpose four independent fuel injection pumps—one for each cylinder—are provided, and the fuel is delivered to the fuel injection valves for the minimum time before they open, and for this reason these pumps themselves are reversible. Moreover, when the engine is reversed, the gear is so arranged that the pumps are thrown out of action one revolution before the exchange of the fuel injection valve cams takes place.

Gebr. Sulzer, Winterthur.—With small high-speed Sulzer engines built between 1905 and 1911, before the system of double-port scavenging, in which the upper port for the supercharge was developed, as described on p. 37, the cylinder heads contain a fuel injection, a starting air, and two scavenging air valves (Figs. 79, 80, p. 80, and Fig. 260).

There are two cam shafts—one for the left-hand scavenging air valves and the other for the fuel injection, starting, and the right-hand scavenging air valves (Fig. 260).

The left-hand cam shaft can be rotated through an angle $\beta = 22°$ with respect to that of the crank shaft by means of a mechanism operated by compressed air, and so the left-hand scavenging air valve is set for the opposite direction of rotation. The spiral wheels on the right-hand vertical shaft are not movable, so that the scavenging air valve actuated by this shaft

is irreversible ; in other words, the scavenging is carried out by both the valves, but the supercharge is given only by that on the left.*

As stated, the cam shaft, which has no angular displacement in relation to the crank shaft when the engine is reversed, operates also the starting air and the fuel injection valves, both of which must, of course, be reversed. The reversing gear is original in design, and is shown diagrammatically in Fig. 261. The lever of each fuel injection and of each starting air valve is actuated by the strap on an eccentric keyed to the cam shaft.

These straps have peripheral profiles of a shape to present two noses b and e for the fuel injection and b_1 and e_1 for the starting air valves. By rotating these straps so that the rollers of the levers are actuated by the noses b and b_1 the engine is set for running in the ahead direction, and by

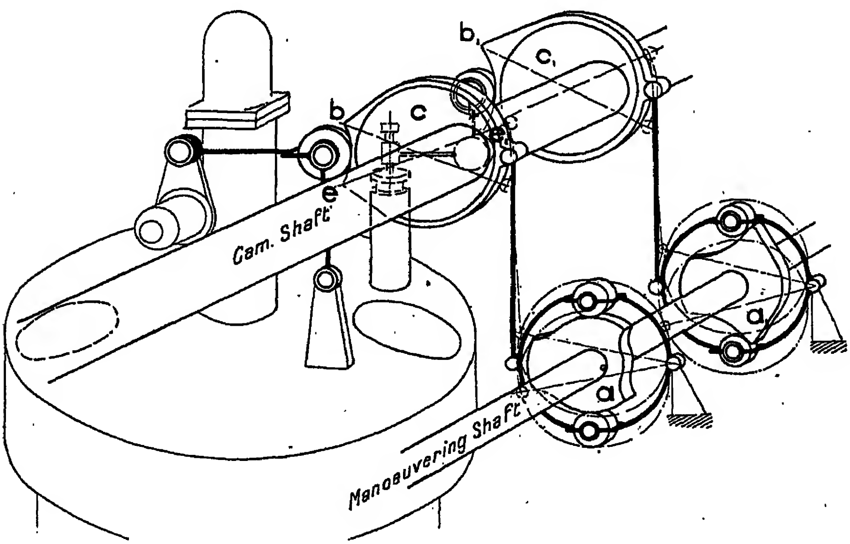


Fig. 261.—Diagram of Early Type of Sulzer Reversing Gear.

rotating the straps in the opposite direction so that the noses e and e_1 are in operation, the engine is set for the astern direction of rotation.

A hand-wheel rotates the manœuvring shaft on which the cams a and a_1 are keyed, which latter serve to set the eccentric collars for the direction of rotation.

The profiles of these cams a a_1 are similar for the same valves of each pair of cylinders, but are different for the two pairs, and so designed that,

* The right-hand scavenging air valve remains open for 80°—*i.e.*, 40° before and 40° after the bottom dead centre—whilst the left-hand one opens also 40° before, but closes 60° after the dead centre. Since the exhaust ports are closed by the piston 50° from the bottom dead centre, 10° of opening of the left-hand scavenging air valve remain for the supercharge.

by rotating the hand-wheel towards the left, the four notches of the sector (Fig. 262) correspond to the following operations :—

(1) Ahead four cylinders on compressed air.
(2) Ahead two cylinders on compressed air and two on fuel oil.
(3) Ahead with two cylinders out of action and two on fuel oil.
(4) Ahead with four cylinders on fuel oil.

Rotating the hand-wheel in the same way towards the right, the same operations are accomplished for astern running.

This system of reversal is not simple, and especially with small engines (*e.g.*, 100 B.H.P. at 400 revolutions per minute) the heads of the cylinders appear crowded with small delicate moving mechanism, but this is the first true system of reversal applied to Diesel engines, and is sure and quick

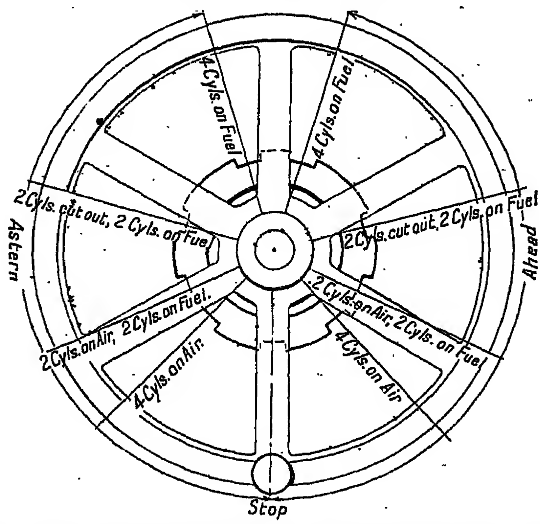


Fig. 262.—Manœuvring Wheel for Valve Gear of Type shown in Fig. 261.

(Messrs. Sulzer have, however, abandoned this system on the introduction of the new method of double-port scavenging).

The author has seen an engine of this type when warm reversed from full speed ahead to full speed astern by turning the hand-wheel as quickly as possible from the right to the left, without even stopping for an instant on the intermediate notches. With one hand on the manœuvring wheel and the other on the control lever of the fuel injection pump full speed between 400 and 450 revolutions per minute is attained more rapidly than is possible with a steam engine.

For engines of the cargo boat type, as built for the M.V. "Monte Penedo" (see Plate VI., facing p. 112, and Fig. 263), reversal is carried out in a totally different way.

Since scavenging is accomplished by a single valve per cylinder, and the scavenging pump is provided with piston valves, in order to change the

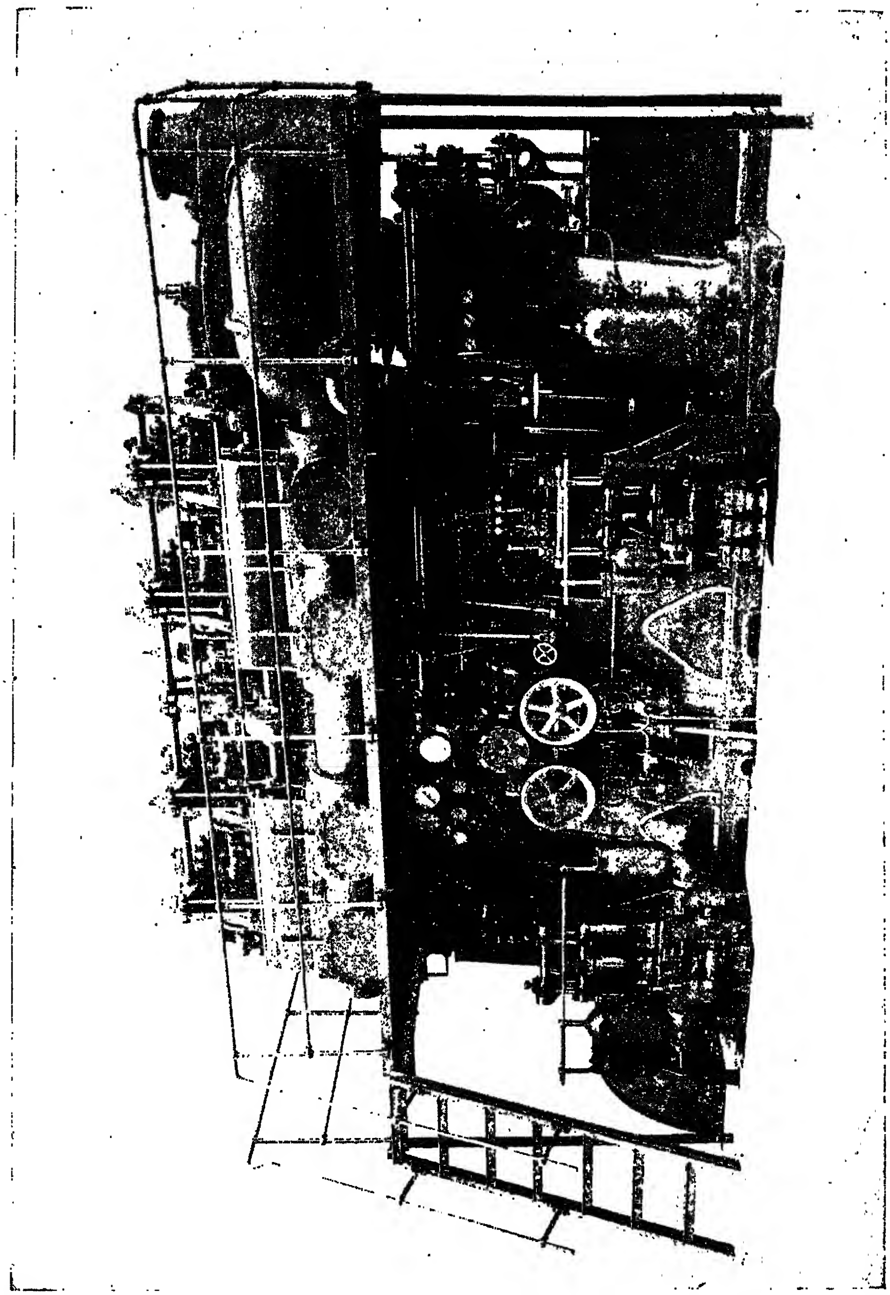


Fig. 263.—One of the two Engines of the M.V. "Monte Penedo" (Sulzer).

direction of rotation of the engine, it is necessary to alter the setting of the starting air, scavenging, and fuel injection valves, and the piston valves of the scavenging pump. These manœuvres are effected by means of two compressed air servo-motors a and a_1 (Plate VI.). The servo-motor a_1 provides for the reversal of the scavenging pump piston valve through an ordinary Stephenson link gear c, and at the same time rotates the cam shaft d through the angle β corresponding to the required timing of the scavenging valve for the opposite direction of rotation. The starting air valve has a double cam n, and the substitution of the cams is also carried out by the same servo-motor, the function of which is thus completely to prepare the engine for the determined direction of rotation.

The other servo-motor a, identical with the first, acts upon the shaft f, which carries the eccentric fulcrums of the starting air and fuel injection valve levers, and causes the engine to run :—

(1) With four cylinders on compressed air.
(2) With two cylinders on compressed air and two on fuel oil.
(3) With two cylinders cut-out and two on fuel oil.
(4) With four cylinders on fuel oil.

An indicator b shows on its quadrant for which of the above methods of running the valve gear is set. Another indicator e, external to the same quadrant and connected to the first servo-motor, shows whether the motor is set for ahead or astern running. By means of a mechanism of the type shown in Fig. 241, p. 170, the position of the fuel injection roller with reference to its cam is corrected after the cam shaft has been rotated through the desired angle. Hence, recapitulating, for the reversal :—

(1) The cam shaft d is rotated by $\beta°$ thus reversing the scavenging valves.

(2) The valve gear of the scavenging pump is reversed by means of the Stephenson link motion.

(3) The new starting air cams are brought into action—*i.e.*, the starting air valves are set $\varepsilon° - \beta°$ out of phase with their original setting.

(4) The position of the rollers of the fuel injection levers is corrected by an angle $\alpha° - \beta°$.

Franco Tosi, Legnano.—With Tosi two-cycle marine engines the scavenging air enters the cylinder through four valves in the cylinder head placed round the fuel injection and starting air valves (Plate VII., facing p. 112).

To reverse the engine, the cam shaft is rotated in the usual way through an angle β, and since $\beta = \alpha$, the scavenging and fuel injection valves are set for the new direction of rotation.

The starting air valve of each cylinder has two cams and two levers, either of which may be brought into action according to the desired direction of rotation. Fig. 264 represents diagrammatically the valve gear for the fuel injection and the starting air valves, and the construction of the same mechanism is shown in Fig. 265.

The eccentric c (Fig. 264) actuates the fuel injection valve lever through the eccentric rod, which is pivoted about a pin in the lower extremity of a link, the upper part of which forms the strap around an eccentric on the manœuvring shaft. The cams a and b lift the starting air valve for ahead and astern running by depressing the lever e, and through this the valve stem.

On the manœuvring shaft are keyed the eccentric fulcrums of the three levers, so that when this shaft is rotated through a certain angle into the

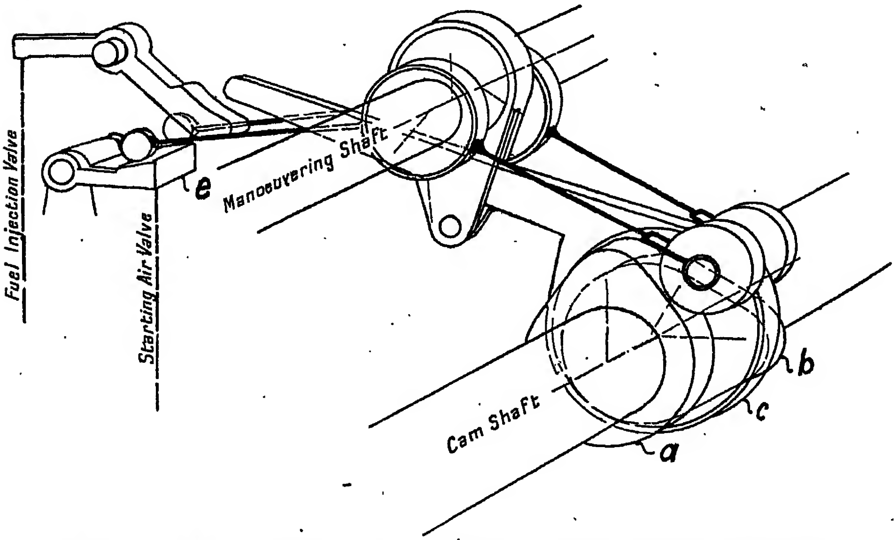


Fig. 264.—Diagram of Valve Gear for Fuel Injection and Starting Air Valves (Tosi).

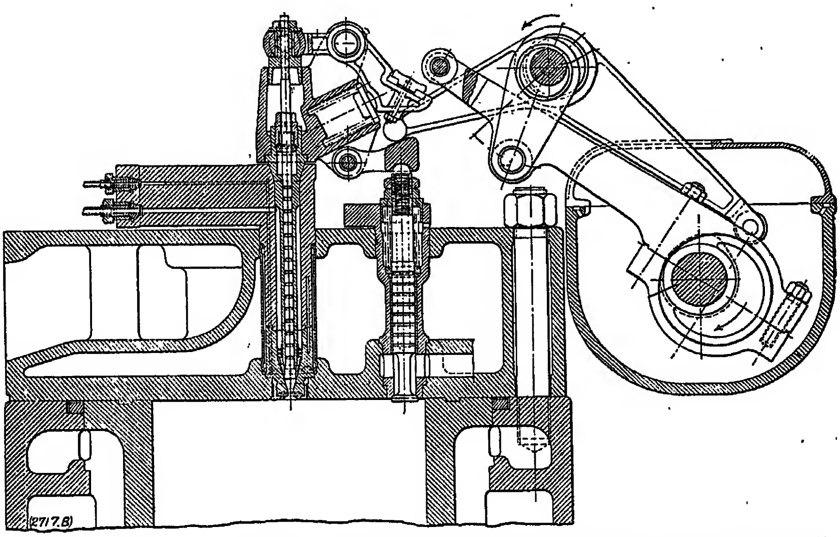


Fig. 265.—Construction of Valve Gear for Fuel Injection and Starting Air Valves (Tosi).

position shown in Fig. 264, neither of the valves are in action. When the manœuvring shaft is rotated in a clockwise direction, first the cam *b* comes

into action and then the fuel injection valve, whilst the lever of the other cam a remains inoperative. By rotating the manœuvring shaft in the opposite direction, after the cam shaft has been rotated through the requisite $\beta°$, first the cam a acts upon the lever e, and then the eccentric c puts the fuel injection valve into action.

This very attractive system has also the advantage of permitting the lift and the time of opening of the fuel injection valve to be varied according to the load on the engine, by suitably adjusting the position of the manœuvring shaft; and so, through the eccentric thereon, the position of the fulcrum about which the rod of the eccentric c oscillates. This condition conduces to economy of compressed air at low speeds. Indeed, when the engine is running at low speeds, if the *angle* of opening of the fuel injection valve is invariable, the valve is lifted for a longer *time* per revolution, and a greater weight of air flows into the cylinder. Since the angle $\frac{\alpha}{2}$, which the bisector of the angle of lift of the valves makes with the vertical, is constant, the angle of lead of the admission, by varying the amplitude of the lift, is also modified when the speed of running and the load are a minimum. It is for this reason that with Tosi engines, the normal speed of which is only 170 revolutions per minute, the condition $\alpha = \beta$ can be satisfied, a condition which is not usually found, even with engines of much higher speeds of revolution.

Reversing Systems apart from the Main Engines.—Reversing systems, apart from those embodied in the main engines, may be classified as follows :—(1) Reversing gears composed of a mechanism of geared wheels, (2) electrical and hydraulic appliances, and (3) propellers with reversible blades.

For the *reversal of the direction of rotation* of the propeller shaft, the system of geared wheels and friction clutches is very satisfactory in its application to engines of limited power and a high speed of revolution, which limit the magnitude of the peripheral forces. Such mechanisms are not suitable for transmitting any considerable torque, and are very rarely to be found applied to Diesel engines.*

Of all the known gears, only one of the most common, as shown in Fig. 266, will be described. A bevel wheel is keyed to the crank shaft and engages with a second, which in its turn drives a third of equal diameter to the first, but free on the propeller shaft. The central lever actuates a sleeve, and causes it to slide along the shaft. This sleeve is connected to the propeller shaft by a feather, and its outer ends form two cones, which serve as friction clutches to connect either of the two gear wheels with the propeller shaft. By moving the lever to the right the propeller rotates in the same direction as the engine, and to the left in the opposite direction thereto. When the lever is in the vertical position the propeller is disconnected from the engine.

The *electrical reversing system* proposed by Del Proposto has many advantages of considerable importance, first amongst which may be cited

* The most powerful marine Diesel installation provided with a reversing gear of this type, known to the author, is that of M.V. "Brioni," which has a Grazer Diesel engine of 120 H.P.

that of its being applicable for considerable powers,* and it has, in fact, been applied, more especially in Russia, to river cargo boats.

In this system a dynamo and an electric motor carried on the propeller shaft are separated by a clutch C (Fig. 267). The dynamo B is connected rigidly to the Diesel engine, whilst the electric motor A always revolves in the same direction as the propeller. For full power running ahead, the clutch is engaged and the engine drives the propeller direct, whilst the electric

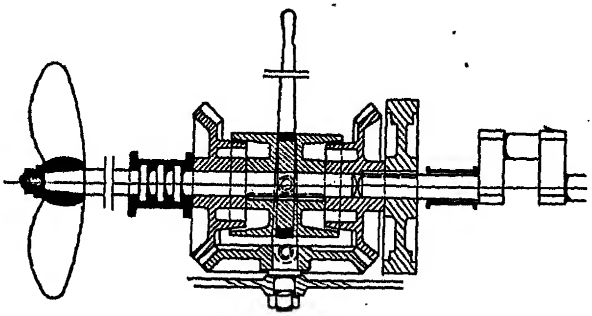


Fig. 266.—Friction Clutch and Bevel Wheel Gear for Reversing Direction of Rotation of Propeller.

machine, being unexcited, revolves freely. For astern running the clutch is put to the "free" position, the dynamo is excited, and delivers its current to the electric motor, which revolves the propeller shaft in a direction of rotation contrary to that of the Diesel engine. By letting the clutch out and exciting the dynamo as desired, and by making use of the reversibility of the electric motor, slow running can be obtained through this medium in both directions of rotation. With this system, the installation, if desired, may be controlled from the bridge, and the Diesel engine may be always running, not only in the same direction of rotation, but at its normal number

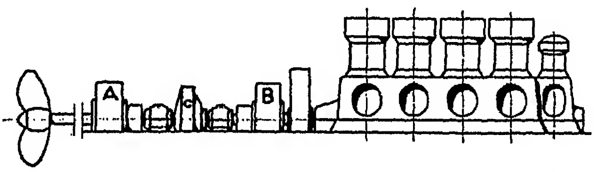


Fig. 267.—Arrangement of Del Proposto System of Electrical Transmission.

of revolutions, automatically regulated by a centrifugal governor, as with land engines. This last condition is one of considerable importance, since one of the difficulties which require to be overcome with marine Diesel engines, especially those working on the four-stroke cycle, is that of obtaining a sufficient degree of flexibility as defined by the ratio of normal to minimum speeds of revolution under continuous working conditions.†

* Supplement to the *Rivista Marittima*, Oct. 1906.

† With marine two-cycle Diesel engines of six cylinders a minimum speed of running of 40 revolutions per minute is attained, and is generally suitable for the most delicate manœuvring of the ship.

Even by installing an electrical manœuvring plant of this type having a power lower than that of the main engine (50 to 70 per cent.), the very considerable advantages of flexibility that are gained may more than counterbalance the obvious defects of increased space, weight and first cost.

In the *Engineer* of the 5th November, 1909,* a description appeared of a machine (the function of which was to reduce the revolutions of the propeller shaft as might be desired, and to provide for the reversal of the drive of a steam turbine), which was installed by the Vulcan firm of Stettin in a small experimental vessel. This device (which consists essentially of two couples of centrifugal turbine hydraulic pumps, one for ahead and the

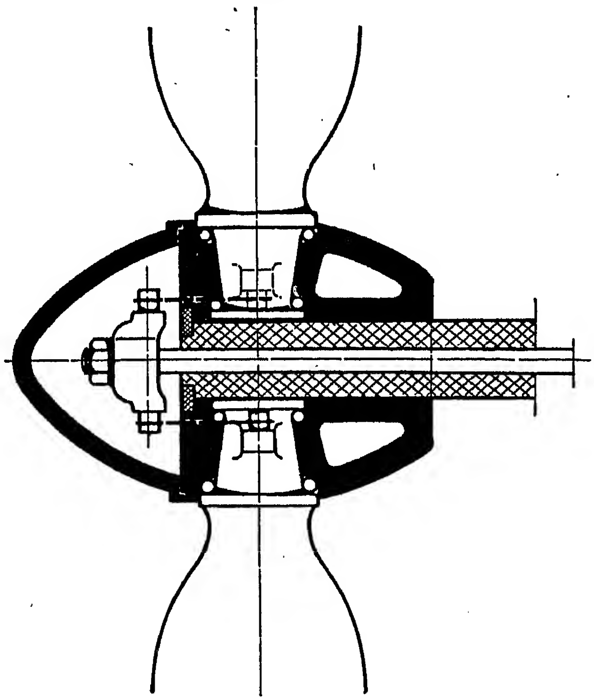


Fig. 268.—Bevis Reversible-bladed Propeller.

other for astern running) is neat and ingenious, and may be applied to any irreversible engine.

Up to the present, however, such a gear has not been installed, so far as is known, for reversing the drive of Diesel engines,† and it is doubtful if it is exactly suitable for this work, in view of the loss of 20 per cent. in the pump turbine group, however well designed and constructed it may be, and the advantages presented by the system (somewhat similar to those of the Del

* See also *Zeitsch. des Ver. deut. Ing.*, 4th December, 1909, and *Rivista Marittima*, 1910.

† It is believed that a transformer of this type is being supplied to the Motor Packet "Pioneer," provided with engines of 1,300 B.H.P., for the Congo State.

Proposto system) are hardly such as to compensate for a loss of power of 20 per cent. in the transformer.*

Reversible Propellers.—Reversible propellers have had considerable success with petrol, paraffin, and heavy oil explosion engined ships, but have the disadvantage of not being applicable for other than small powers.

It is obvious that the mechanism for varying the angle of the blades is awkwardly placed in the boss of the propeller, and that the blades, being movable at the point where they are most highly stressed, cannot be so rigidly connected to the boss of the propeller as when they are bolted to the boss or cast solid with it, and that finally the reversing mechanism, the certainty of action of which is so necessary, is completely inaccessible when the vessel is in the water. On the other hand, it may be stated that no

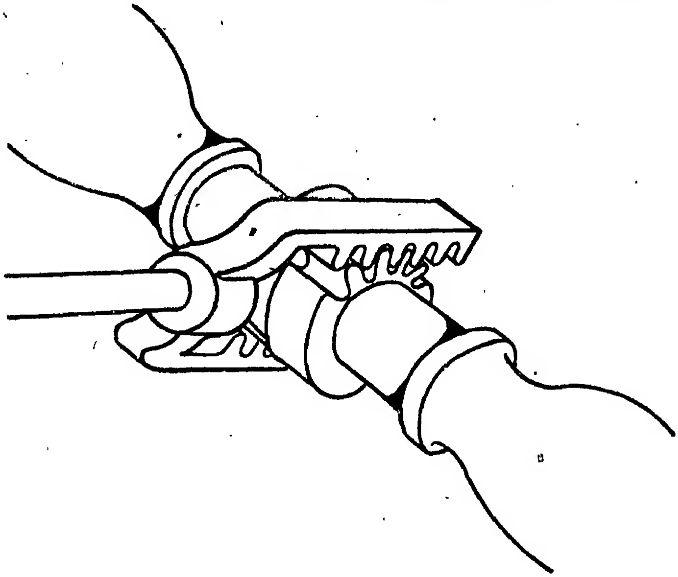


Fig. 269.—Diagram of Mechanism of Weihe Reversible-bladed Propeller.

other device can compete with this in lightness, simplicity, first cost and easy working.

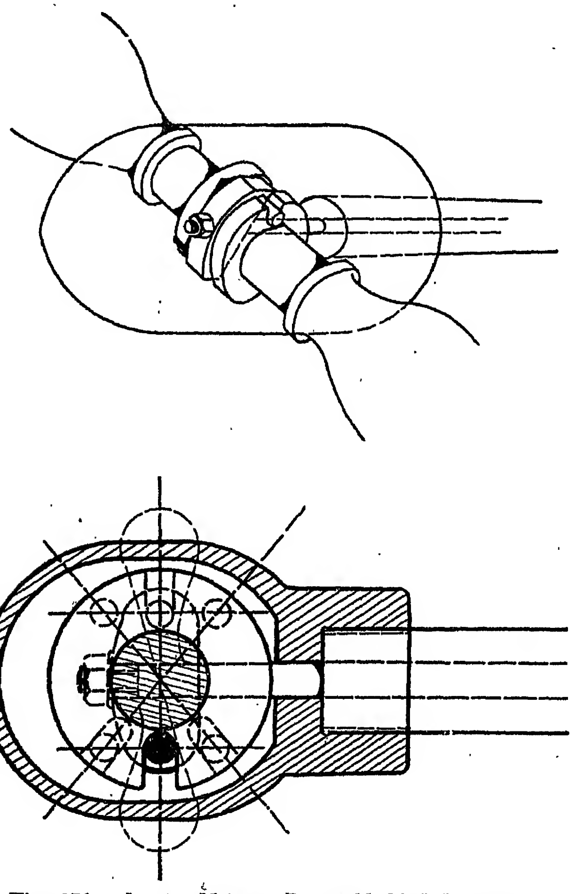
The figures which follow show some of the better-known devices for reversible propellers.

In Fig. 268 is shown the Bevis propeller, which was the first to be applied to installations of relative importance. It has been applied for the transmission of several hundred H.P. in British auxiliary engined corvettes, not

* This gear is known as the Föttinger transformer, and in its application to steam turbines has advantages which have little force in considering its feasibility for the reversal of Diesel engine drives. It permits of running the turbine at the maximum speed of revolution and the propeller slowly, that both may work under the conditions of maximum efficiency, and so the loss of power in the transformer and the extra weight due to the same are compensated for by the reduction in dimensions of the ahead turbine and the elimination of the astern turbine.

only to reverse the direction of motion, but also to reduce the resistance of the propeller when the ship was proceeding under sail. By feathering the blades of the propeller, the well, which was formerly necessary for lifting the propeller out of the water when sails exclusively were being used, is dispensed with.*

The propeller shaft is hollow, and through it is led a shaft terminating



Figs. 270 and 271.—Meissner Reversible-bladed Propeller.

in a crosshead. By means of two connecting-rods this crosshead rotates the blades of the propeller through cranks on these latter.

Fig. 269 shows the action of the Weihe propeller successfully applied by G. F. Deutz for powers up to 120 B.H.P.

The Meissner propeller (Figs. 270 and 271) is similar to this latter, and is frequently used, more especially for propellers with three blades.

* Achenbach, *Die Schiffsschraube*, part ii.

In the construction of reversible-bladed propellers, the shape of the blade has to be carefully designed to compensate as far as possible for the thrusts which would tend to rotate them around their axes.

Perfect compensation for all speeds and blade angles is impossible, but it is necessary to endeavour to approach as far as possible the ideal condition, in order to reduce the difficulty of reversing the blades and diminish the strains set up in the internal mechanism.

Reversible propellers are generally of bronze or of steel, whilst those of large diameter, subject to considerable stresses, are of special bronze or chrome-nickel steel. The largest propeller of this type constructed up to the present is that of the well-known "Theodor Zeize," of Altona, for the German Navy, to transmit 900 B.H.P. The diameter is 1·40 metres,* and the peripheral speed 40 metres per second. The pins of the blades are lubricated by a type of Stauffer lubricator enclosed in the boss, containing sufficient lubrication for several hundred reversals. The mechanism is such that the internal manœuvring shaft at each reversal presses out a small quantity of lubricant. The lubricators are charged each time the ship is in dry dock. Complete reversal is accomplished in twenty seconds.

* *Zeitsch. des Ver. deut. Ing.*, 14th September, 1912; W. Helling, *Die Umsteuerschrauben für grosse Leistungen.*

CHAPTER XII.

FUEL INJECTION PUMP AND FUEL REGULATION.

In the Diesel engine the fuel injection pump delivers the fuel to the fuel injection valve in the cylinder head, and must, therefore, overcome the pressure (from 45 to 75 atmospheres—*i.e.*, 640 to 1,100 lbs. per square inch) in the valve casing, due to the fuel injection air. The pump also regulates the running of the engine, and, therefore, the fuel delivered at every pump stroke needs to be exactly that quantity necessary for one combustion stroke of the engine, at the load under which it is working at the moment.

Since with an exceedingly small delivery, variable through wide limits, the pump has to overcome this very high pressure, it must be strong and have, at the same time, very delicate mechanism. To fulfil these requirements the greatest care must be exercised in the design, and only the finest workmanship will suffice for the construction.

The piston is always of the plunger type, of steel; the valves are of bronze, cast iron, or steel, with conical seatings, one suction and one, or two in series, for delivery, loaded with light springs, and so disposed as to be readily accessible for examination, cleaning and grinding-in. The joints of the copper delivery pipes are made with conical connections, as is shown in Fig. 185, p. 133.

The body of the pump is of cast iron, of massive construction; the plunger and other moving parts working under pressure have carefully packed glands. A good packing for this purpose is made of asbestos cord, greased with tallow and black lead.

The manner in which the delivery of the pump is controlled by the governor, in accordance with the load on the engine, may now be considered.

The first idea which suggests itself for the solution of this problem is that of varying the stroke of the plunger, modifying the throw of the actuating eccentric by means of an axial governor. But it is obvious that the pump would not be able, at the lowest powers and with the engine running light, to deal satisfactorily with the minute quantities of the dense and viscous fuels employed.

A very small bubble of air in the pump chamber would be sufficient to stop the action of the pump. The plunger in its very small motion would merely compress and expand the bubble of air without raising the valves.

On the other hand, since the pressure which the plunger has to overcome is very considerable (45 to 75 atmospheres—*i.e.*, 640 to 1,100 lbs. per square inch), a gear to give a variable stroke, and one that would not cause a large reaction on the governor, making it oscillate with every stroke of the plunger, would be difficult to design.*

* These difficulties are subject to the condition that the pump has to overcome the injection air pressure. For those types of fuel injection valves in which this is not the case (see p. 137, and following), it is possible to adopt, as indeed are adopted, systems using a variable pump stroke.

Thus the problem has to be solved in such a way that the regulation is not obtained by any alteration of the plunger stroke, and that a quantity of oil corresponding to the whole pump cylinder volume passes the suction valve each suction stroke. For this reason, the fuel injection pumps of

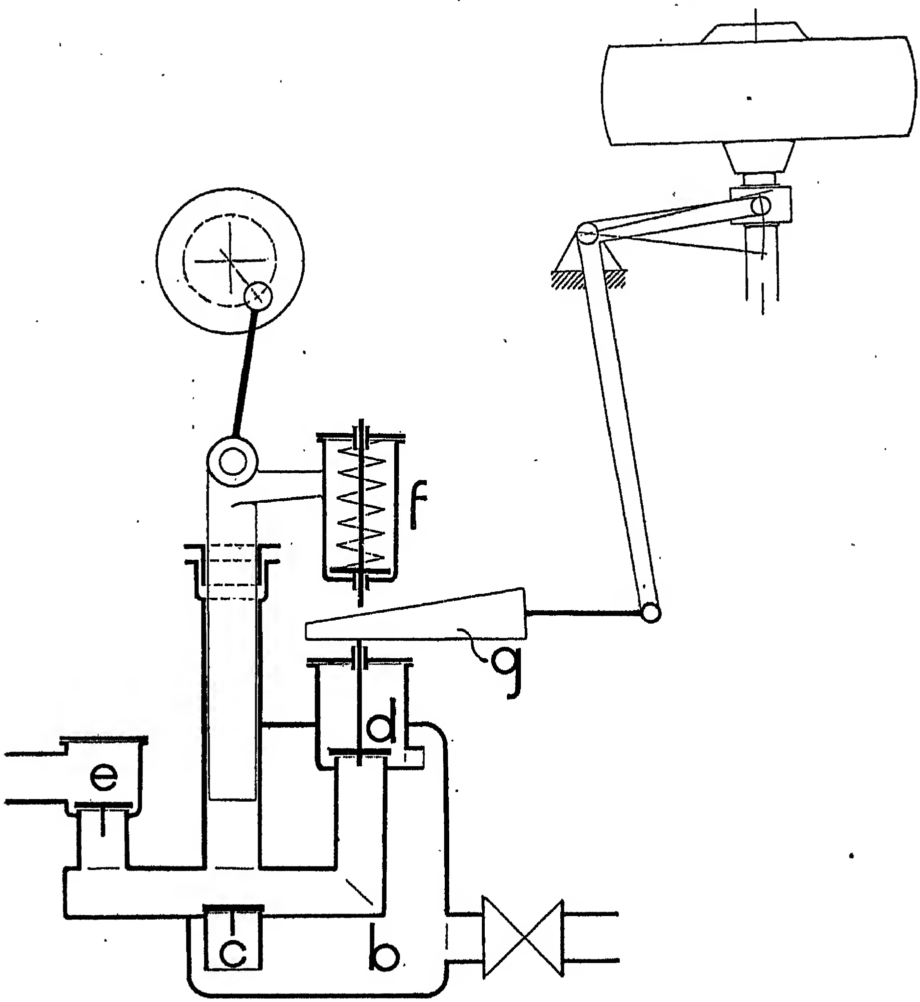


Fig. 272.—Diagram of one of the Earliest Designs of Fuel Injection Pump showing Governing Device.

Diesel engines draw a quantity of oil in excess of that required, of which a part only goes to the fuel injection valve, the excess passing back through the suction valve during a part of the delivery stroke.

In calculating the pump cylinder volume, a consumption of fuel of from

600 to 900 grammes (1·32 to 2 lbs.) per B.H.P. per hour is always assumed. Regulation takes place at all loads, or, in other words, even when the engine is working at an overload, only a fraction of the delivery from the pump goes to the fuel injection valves on the engine cylinders.

The actual arrangements designed to achieve this result are many, but all are similar in their fundamental conception. The oldest arrangement, now abandoned, is shown in Fig. 272.

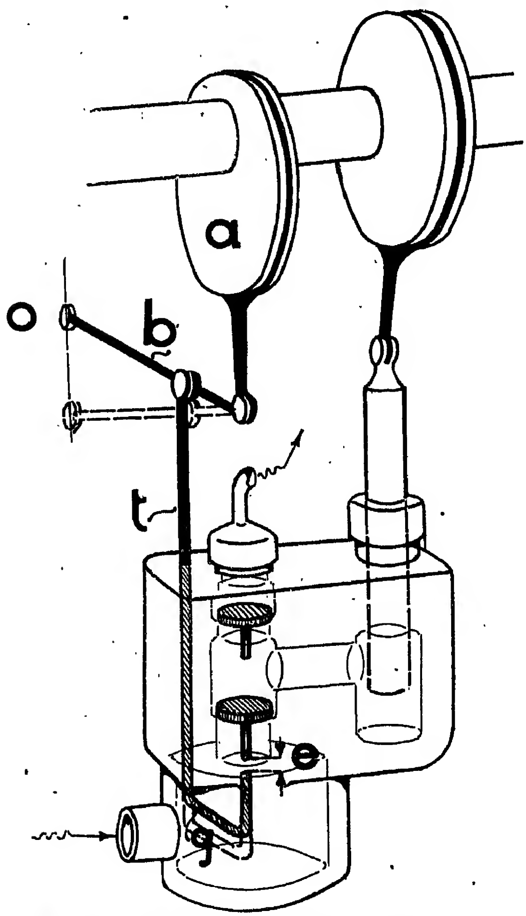


Fig. 273.—Diagram of Typical Fuel Injection Pump.

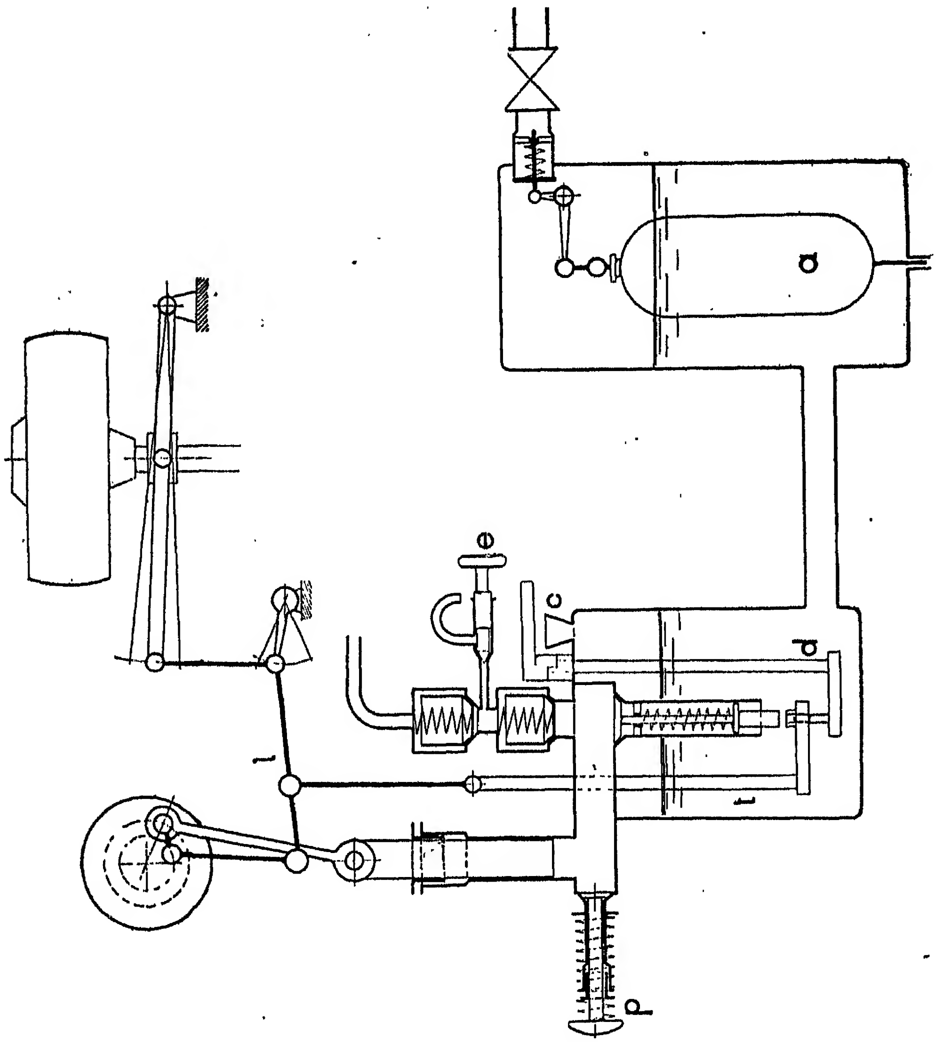
The pump plunger on its upward stroke draws the fuel oil contained in *b* through the suction valve *c*. If, when the plunger descends, the valve *d* be held on its seating, all the fuel will go to the fuel injection valve through the delivery valve *e*; if instead the valve *d* be left free, the fuel will raise the valve *d* in preference to the valve *e* (which is under the pressure of the fuel injection air), and will return to the suction chamber *b*.

If, then, for a certain part of the stroke the valve *d* be left free, and for the remaining part held in its seat, of the total amount of fuel dealt with by the pump, only the fraction delivered during this latter period will find its way to the fuel injection valve. It is in this way that the operations of the pump (as clearly shown in Fig. 272) are controlled by the spring plunger *f* and the wedge piece *g*, which latter is moved by the governor.

Fuel injection pumps, as fitted to modern Diesel engines, may be grouped, according to their appearance and mechanical details, into four or five types, which will be described, but the fundamental idea underlying their regulation is the same in all cases, and is as shown in Fig. 273.

An eccentric *a* causes a lever *b* to oscillate about the point *o*. To the lever *b* is connected a link *t*, ending in a hook *g* under the suction valve *e* of the pump. The stroke of the eccentric and the greatest distance *e* between

the highest point of the hook and the lowest point of the suction valve are such that during the time the eccentric revolves through a certain angle on either side of its upper dead point, the hook keeps the suction valve raised. If this happens during the delivery stroke of the pump plunger, the fuel



drawn into the pump will not be delivered to the delivery pipe until the hook has left the suction valve free, and the latter has returned to its seat.

By displacing the point o vertically, the initial distance e, and in consequence also, the amount and the duration of lift of the suction valve due to the effect of the hook, are varied, and thus the delivery of the pump is also varied.

Figs. 274 to 286 represent some of the more general arrangements of the various parts.

In the type shown in Fig. 274 the fuel flows from the reservoir to a float chamber, from which it passes to the suction chamber of the pump through a tube of sufficient section for easy flow, even in the coldest weather when the fuel is very viscous. This suction chamber is in communication with the atmosphere through a vent *c*,* and the valve operated by the float *a* maintains the fuel at a constant level.

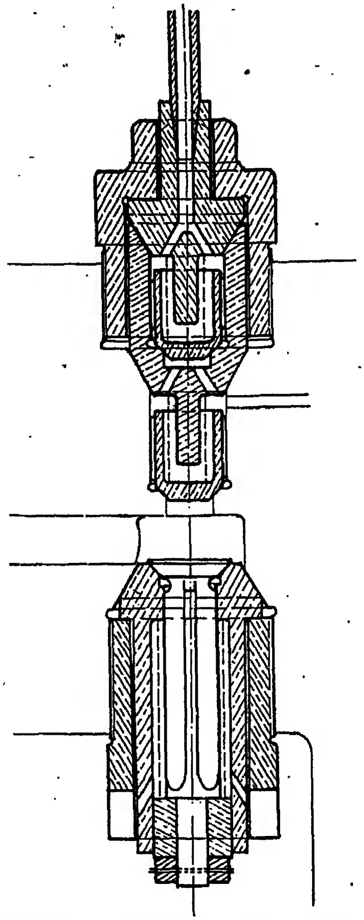


Fig. 275.—Arrangement of Fuel Pump Suction and Delivery Valves.

The pump has a plunger piston, one suction valve, and two cup-shaped delivery valves in series. The presence of the second delivery valve makes it possible to fit between the two a test valve *e* with a conical seating, by opening which, even when the engine is running and the fuel injection valve is under pressure, it is possible to test the regularity of the pump discharge.

Fig. 275 shows the arrangement of these valves, and Fig. 276 that of the pump.

The hand-controlled plunger *p* (Fig. 274) permits of the pump and the piping to the fuel injection valve being filled when, after an overhaul, they may have been emptied, and serves thus to ensure a small quantity of fuel being present in the fuel injection valve before the engine is started. This plunger has an end in the form of a valve, in order to avoid the necessity of carefully packing the gland, an operation of considerable difficulty in view of the small diameter of the plunger. In fact with this form of combined plunger and valve, there is no necessity for a packed gland, since the high pressure is a guarantee that the valve will be held tight on its seat.

The hook *d*, Fig. 274, connected to a hand lever, permits of the suction valve being held open, thus stopping the action of the pump, and in consequence, the running of the engine.

In addition to the eccentric working the plunger, there is a second keyed on to the shaft at an angle generally different from that of the first. The

* Through the vent *c* a little paraffin can be introduced at starting, to make the fuel more fluid.

latter regulates the fuel delivery by means of the lever *l* and the link *t*, ending in another hook situated under the suction valve; this hook is raised or lowered according to the position of the governor, which, by modifying

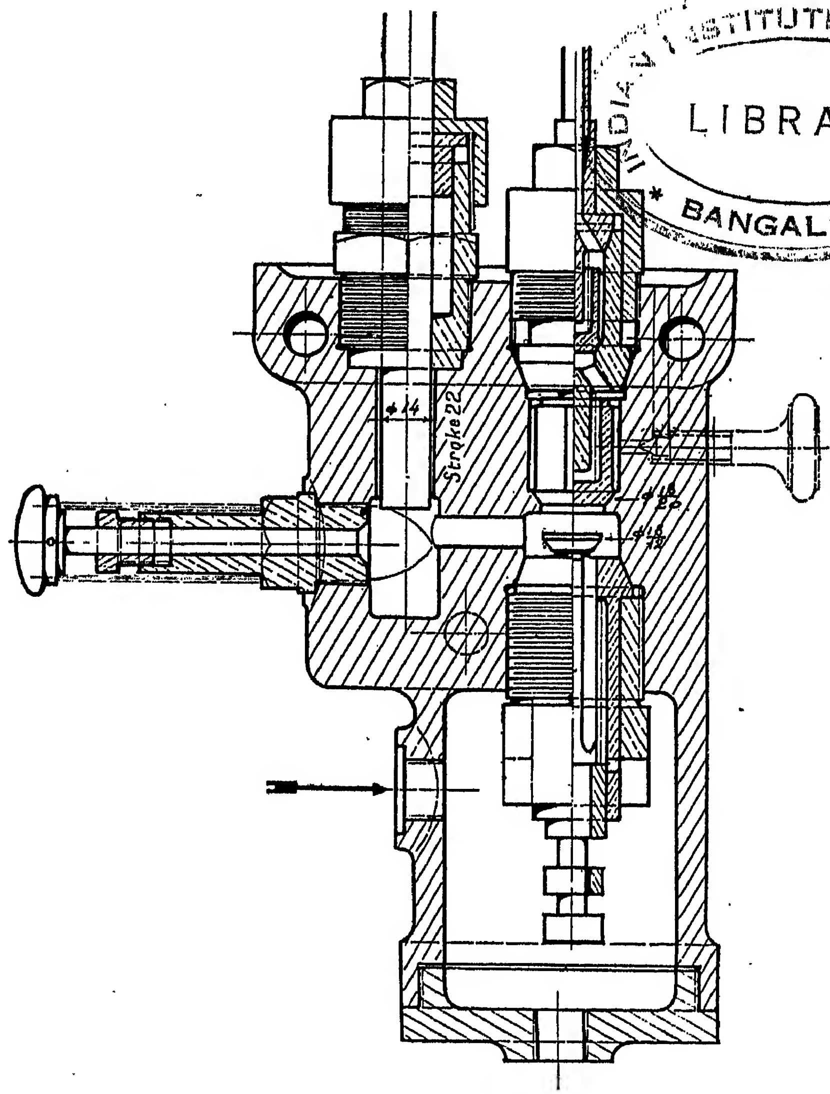
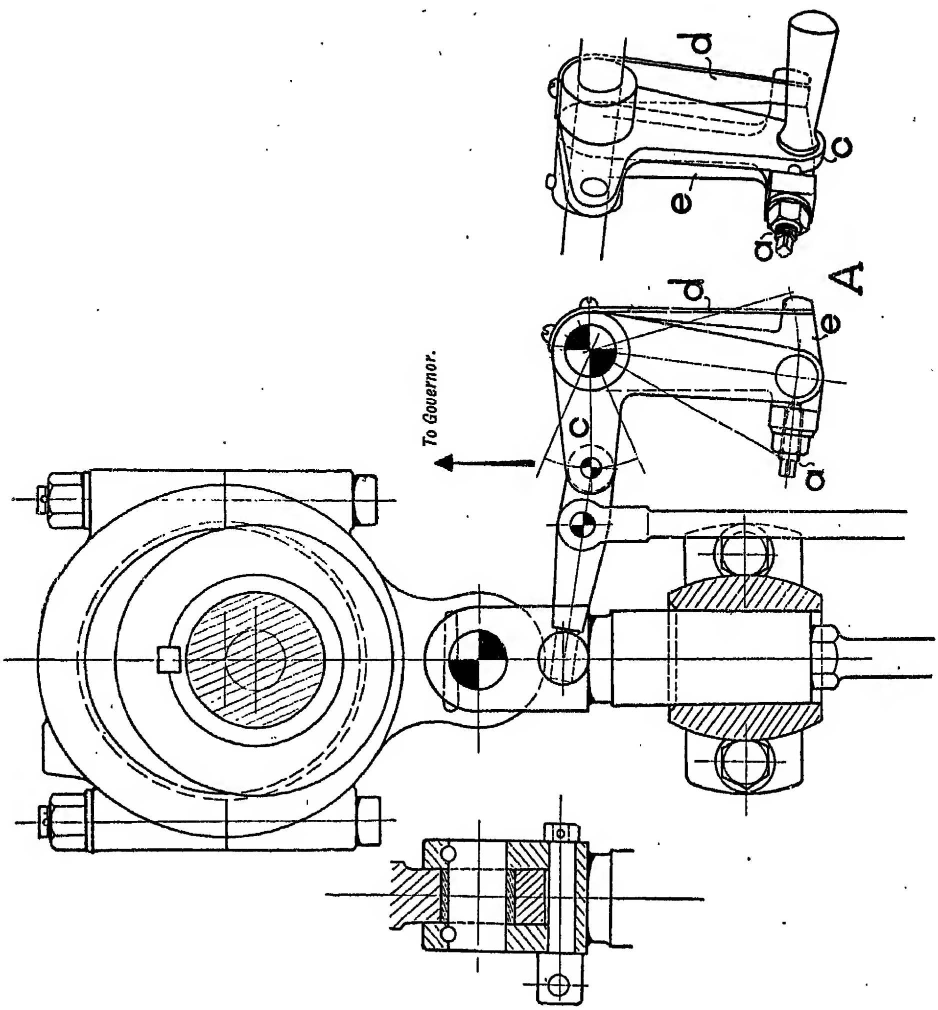


Fig. 276.—Arrangement of Pump shown in Fig. 274.

the initial position of the movement of the link *t* and of the hook (the stroke being constant), causes the hook to hold the suction valve open through a longer or shorter part of the delivery stroke.

As will be shown later, the two eccentrics of the pump may be keyed on at the same angle, in which case one of them may be eliminated, as shown in Figs. 277 to 279.

Fig. 277 shows also a type of arrangement (A) which is to be found in nearly all fuel pumps. The bell crank lever *c* is not rigidly fixed to the



shaft worked by the governor; to this shaft, on the other hand, is keyed the lever *e*, to which is fastened the spring *d*, keeping *c* in contact with the adjusting screw *a*.

By turning this screw, the position of the lever *c* is changed with respect to that of the governor, and the initial distance between the hook and the

suction valve is altered. In other words, the screw *a* permits the delivery of the pump to be varied for a given position of the governor.

By pushing the handle of the lever *c*, overcoming the pressure of the spring *d*, it is possible, by lowering the hook so much that it is not permitted to touch the suction valve, to obtain the full output of the pump on starting the engine.

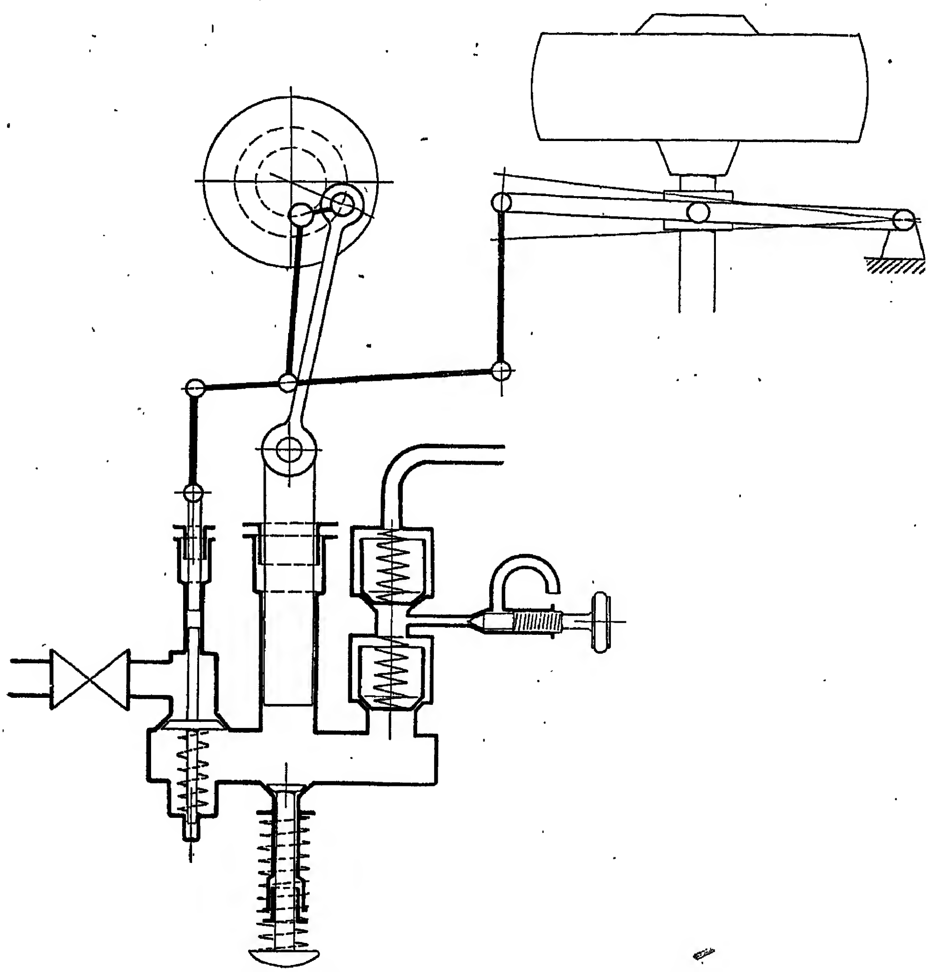


Fig. 280.—Diagram of Fuel Pump Regulation (Langen & Wolf, Grazer, etc.).

The design of pump represented in Fig. 280 is derived directly from the one just described; its general action and the regulation are identical, and all the details of the pump previously described are to be found—the delivery valves between which the test plug is fitted, the hand pump for filling the fuel piping, etc. In this case, however, the suction, instead

of being taken from a chamber at atmospheric pressure, is under the hydrostatic pressure of the fuel reservoir, and so the float chamber is dispensed with.

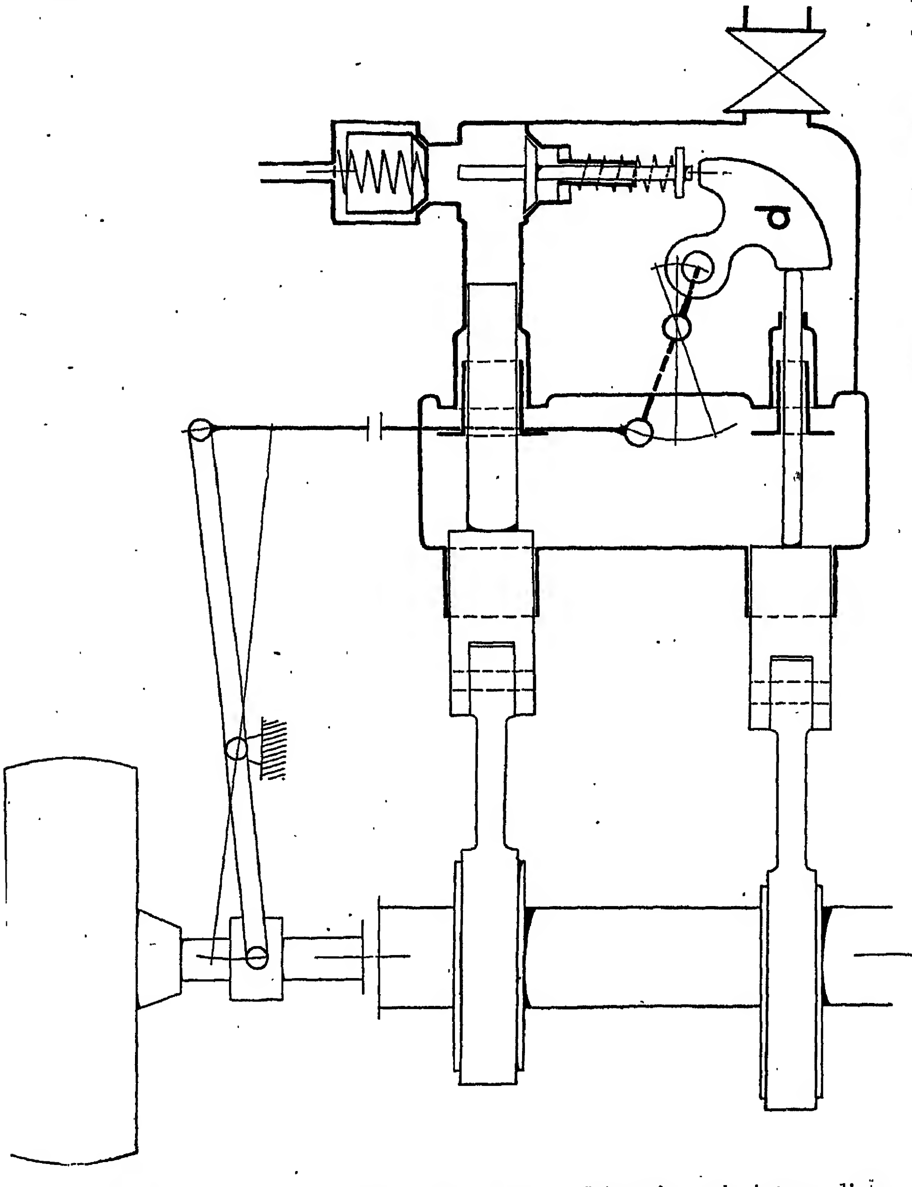


Fig. 281.—Diagram of Horizontal Fuel Pump and its Regulation (Tosi, Sulzer, A.B.D.M. Stockholm, Carels, S.L.M. Winterthur).

The following designs of horizontal pumps are driven from the intermediate vertical shaft, whilst those hitherto described take their motion from the

horizontal cam shaft. The cam shaft, in four-cycle engines, revolves at half the speed of the crank shaft, whilst the intermediate vertical shaft generally revolves at the same speed as the engine, to accommodate the governor, since a lower number of revolutions of the intermediate vertical shaft would necessitate a large-sized governor to give the requisite energy for controlling the valves. For this reason, when they take their motion

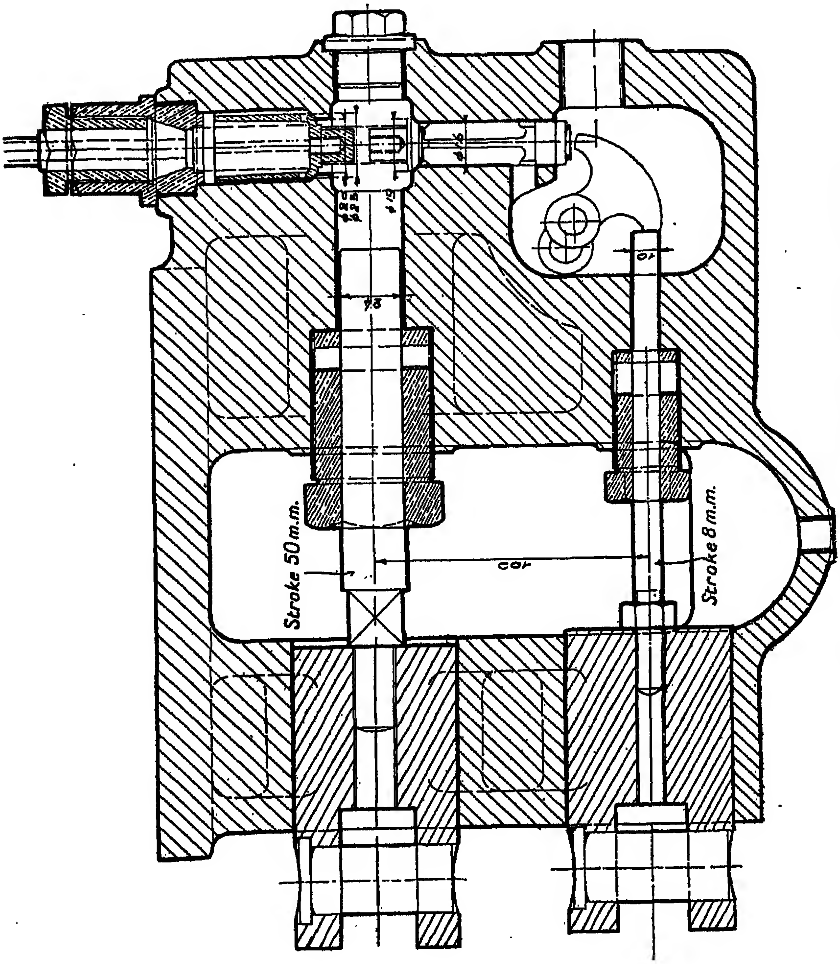


Fig. 282.—Arrangement of Horizontal Fuel Pump shown in Fig. 281.

from the vertical shaft, with four-cycle engines, the fuel injection pumps have two delivery strokes for each combustion stroke of the engine.

Figs. 281 and 282 represent a pump of this type, having two eccentrics, one operating the plunger, the other controlling the suction valve. Control of the suction valve is obtained in the usual way by holding it open, through the medium of the angle piece *d* (Fig. 281), during a part of the delivery

stroke. The pin about which this angle piece pivots is not fixed directly to the pump casing, but to an internal lever, connected externally to the governor. The motion of the piece *d* is, therefore, practically constant, but for different positions of the governor, which raises or lowers its fulcrum, the period of the delivery stroke, during which the suction valve remains open, is varied.

An arrangement which permits of the suction valve being kept open throughout the whole revolution serves to stop the engine.

In addition, it is possible to dispense with the small hand pump by utilising this gear to lift the suction valve to such an extent that it in turn raises the delivery valves off their seats and allows the fuel to flow into the piping under its own head. By opening a small vent in the fuel piping near the fuel injection valve, air is released and the fuel flows freely. Figs. 283, 284, and 285 show the arrangement described. In I is seen the running

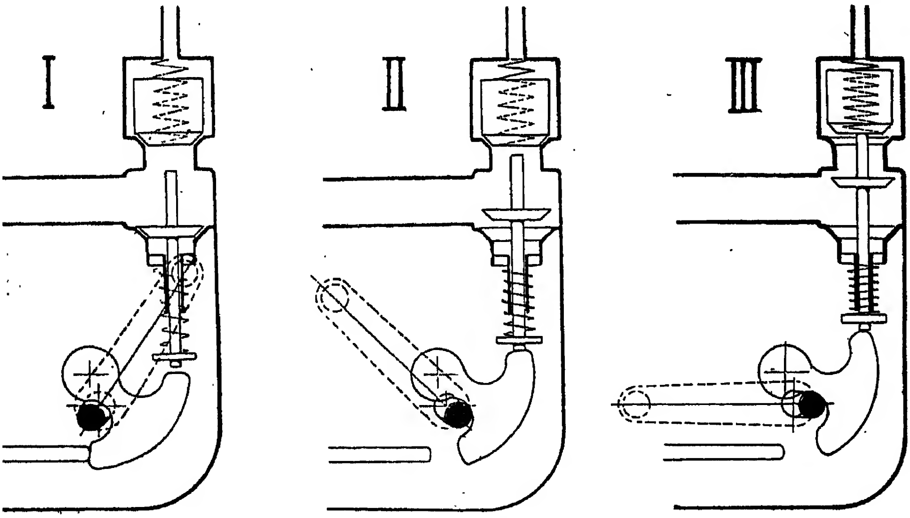


Fig. 283.—Running Position.

Fig. 284.—Stop.

Fig. 285.—Flooding Fuel Piping.

Figs. 283 to 285.—Diagrams illustrating Operations of device for stopping the Engine or flooding Fuel Piping previous to starting with Pump shown in Figs. 281 and 282.

position, in II that for "stop," and in III that for flooding the fuel piping.

For this type of pump a test plug similar to that shown at *e* (Fig. 274) is necessary, and in place of fitting a separate valve for this purpose, the vent screw above mentioned, which serves to open communication with the atmosphere on the fuel injection piping, is frequently made use of. Between this test plug and the fuel injection valve a non-return valve is fitted to prevent the fuel injection air from escaping when the test plug is opened.

A modification of this pump, as regards the method adopted for regulation, is shown in Fig. 286.

The fulcrum of the angle lever *d* (Fig. 286) is fixed, and the variation

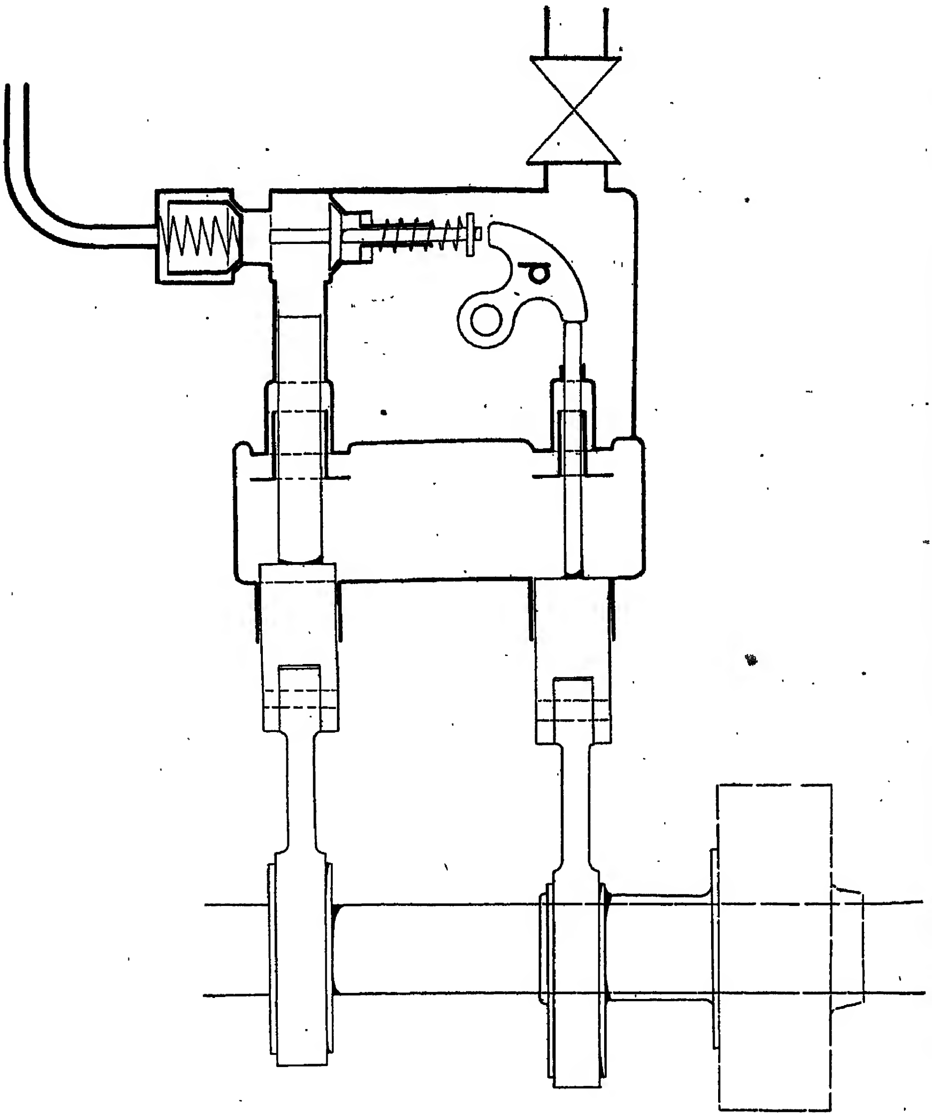


Fig. 286.—Diagram of Fuel Pump in which Regulation is obtained by varying the Angle of Lead of the

of the quantity of the fuel supply is obtained by varying the angle of lead of the eccentric operating the suction valve, relatively to that of the eccentric actuating the plunger.

This variation is obtained by the action upon the distributing eccentric of an axial governor of the type represented diagrammatically in plan in Fig. 287.

As already stated, the design of pump shown in Fig. 272 is now abandoned, whilst the types illustrated by Figs. 273 to 286 are all widely used with satisfactory results in practice.

The arrangement shown in Fig. 274 is of the type adopted by Messrs. Langen & Wolf, M.A.N., G. F. Deutz, Harlé, Güldner, Savoia, etc.; that in Fig. 280 is met with in engines of Langen & Wolf, Grazer, etc., and those shown in Figs. 281 and 286 indicate the practice of Tosi, Sulzer, A.B.D.M. Stockholm, Carels, S.L.M. Winterthur, etc.

The designs most generally adopted may be classified into vertical and horizontal types, according as the motion is received from the horizontal cam shaft or from the intermediate vertical shaft. Many characteristics are

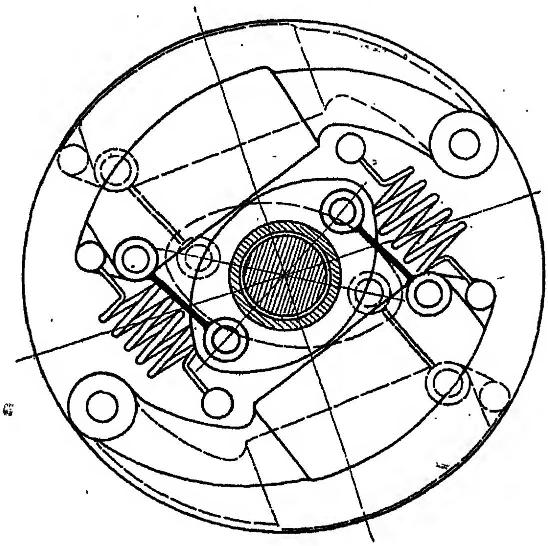


Fig. 287.—Axial Governor for Fuel Regulation.

common to these two types, and although generally not dependent, as regards the main structural features, upon whether the driving shaft be vertical or horizontal, yet it so happens that this latter consideration serves well to differentiate the design.

As an example of the foregoing, it will be found that when the fuel injection pump is driven from the vertical shaft, one pump usually serves to supply all the cylinders, whereas if driven from the horizontal cam shaft separate pumps are generally provided for each cylinder.

In the former case, at some point in the fuel delivery piping, a distributor is fitted, from which branch as many pipes as there are fuel injection valves to be supplied. As the total delivery of the pump is to be divided between them in equal parts, there is at the connection of each branch pipe a steel diaphragm, with a minute hole calibrated by trial, in order to com-

pensate for the different losses of pressure to which the oil is subjected in reaching the various cylinders.

In engines of recent design, a separate pump is frequently adopted for each main cylinder to dispense with the distributor, and the pumps, cast in one piece, have their plungers driven by a single eccentric working a cross-head to which they are all attached. This type of multiple pump, although making it easier to ensure an equal division of power between the various cylinders than does the use of a single pump and distributor, does not, as might at first appear, fulfil the idea which inspires those constructors who adopt one independent pump for each main cylinder as the solution for ideal fuel regulation.

These independent pumps have their eccentrics keyed on at different angles, corresponding to the angles of the cranks of their respective cylinders, that each may deliver fuel to the injection valve it supplies, just before the valve opens. Thus, directly the governor has moved under the influence of change of load, the cylinder which first operates on the combustion stroke will receive fuel from the pump, regulated by the governor conforming to the new load. On the other hand, in cases where there is a single pump with a distributor, or—what comes to be the same thing—a multiple pump with a single-driving eccentric, should the load vary when the fuel injection valves

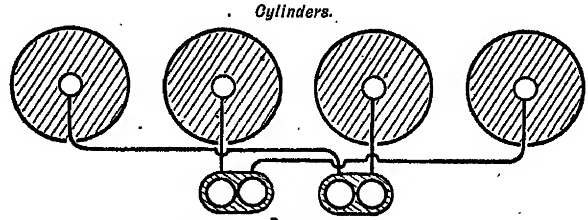


Fig. 288.—Arrangement of Pipes from Four Fuel Pumps operated in pairs by Two Eccentrics.

have already received their supply of fuel, as regulated by the position of the governor before the change, it follows that all the cylinders will have one combustion stroke to perform corresponding to a power in conformity with the previous instead of with the actual load. In this way, the correct fuel supply will not be established before the succeeding series of combustion strokes.

This consideration is, in practice, of much less importance than might appear, on account either of the rapid succession of cycles, of the relative sluggishness of movement of the governor, or of the momentum of the flywheel. It is also made of less account with four-cycle engines, owing to the fact that the horizontal pumps, taking their motion from the vertical shaft, give, as already stated, two delivery strokes for every combustion stroke of the engine, so that at the second stroke they may have come under the influence of the governor, whereby the total supply of fuel is more proportionate to the load.

Pumps with vertical plungers are also sometimes made of the multiple type. For example, in engines with four cylinders the four pumps may be united in pairs, actuated only by two eccentrics, but in this case the delivery pipes will be arranged as shown in Fig. 288, so that, although the pumps are

not exactly in phase with the cylinders, a sufficient approximation is obtained.

It has been seen in several types of pumps, especially when horizontal, that the hand-actuated plunger is omitted and in its stead the test-plug is fitted close to the fuel injection valve, so that the fuel, prior to starting, is in a position rapidly to reach this valve when the engine is put in motion. With the hand pump, besides being able to deliver some fuel into the fuel injection valve before starting the engine, there is the further advantage, owing to the hand pump valves being the same as those of the main fuel pump, that it is possible to ascertain if the fuel injection pump is working correctly. This reduces the probability of a failure to start.

In those pumps provided with a float for regulating the fuel level, in which the suction chamber is in communication with the atmosphere, it has already been pointed out that, to facilitate starting, a little paraffin may be intro-

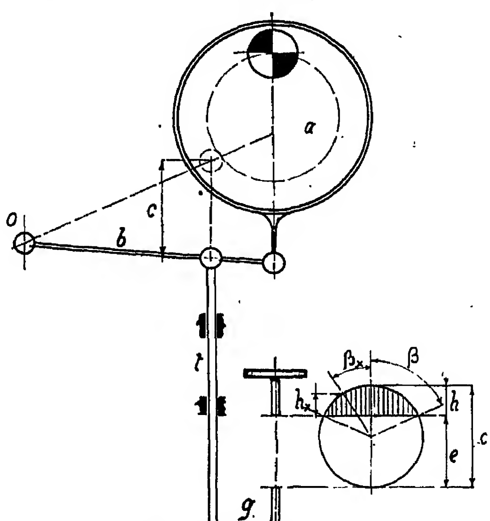


Fig. 289.—Fuel Distributing Mechanism with Eccentric at its Lowest Position.

duced through the vent c (Fig. 274, p. 197). In all other types, it is desirable that the fuel cock to the pump should be of the three-way type, communicating with the heavy oil tank, and with another containing a lighter and more fluid fuel. If the precaution is taken to run the engine with paraffin for the last minute or so previous to stopping, the pump and the delivery pipes will be charged with this fuel, and the re-starting of the engine will be greatly facilitated.

Calculation of Fuel Pumps.—The diagrams and calculations for the fuel injection pump, and for its regulation, may now be considered.*

With further reference to the arrangement (Fig. 273), it has been shown that the hook g, moved by the distributing eccentric a, must lift the suction

* The treatise here given was first published by the author in *Il Polytecnico*, No. 8, 1912, "Variable Delivery Pumps."

valve of the pump, and keep it raised for a fraction of the delivery stroke, in order to permit of the return of a portion of the fuel to the suction side.

Thus, only when the hook g has descended far enough to leave this valve on its seating, will the useful part of the delivery stroke commence.

In the diagram (Fig. 289) the distributing mechanism is shown in the position where the eccentric is at its lower dead centre. Supposing the governor to be in a given position—that is to say, with the point O fixed in space—e is the greatest distance between those points of the hook and of the suction valve which come into contact the one with the other.

A circle of diameter equal to the stroke c of the hook is drawn (on the right of Fig. 289) tangential to the horizontal straight line passing through the extremity of the hook, and is cut by another straight line drawn horizontally through the extremity of the valve, when the latter is resting on its seat.

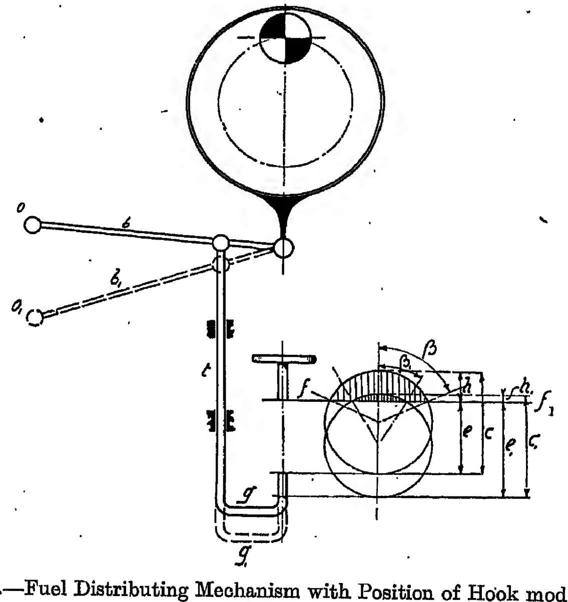


Fig. 290.—Fuel Distributing Mechanism with Position of Hook modified by action of the Governor.

This gives the whole diagram of the action of the distributing system (leaving out of account obliquities of the connecting links).

To each angle β_x of the displacement of the eccentric from its bottom dead centre, corresponds a lift h_x of the valve; 2β is the angle travelled by the eccentric while the suction valve is subject to the action of the hook; $360°-2\beta$ that during which it is free; and h the maximum lift of the valve.

If, due to the action of the governor, the point o is displaced to o_1 (Fig. 290), the stroke c of the hook will remain unchanged, but the value of the greatest distance e will become e_1, and correspondingly the maximum lift h will become h_1, because the circle, of diameter $c = e + h$, will have been lowered with respect to the straight line $f f_1$ passing through the extremity of the valve, the latter remaining fixed in space.

With the displacement of the point o, the lifts h_x are varied in accordance with any angle β_x, and the angle 2β, during which the suction valve was raised by the hook, will be changed to $2\beta_1$.

The deductions will be equally true if, instead of considering the displacement of the circle, of diameter c, with respect to the straight line ff_1 fixed in space, that of the straight line is considered with respect to the circle maintained fixed.

The positions f_1f_1', f_2f_2', f_3f_3', etc., of the secant (Fig. 291) correspond to the various positions of the point o (Fig. 290). The maximum distances e_1, e_2, e_3, etc.; the maximum lifts h_1, h_2, h_3, etc.; and the angles $2\beta_1$, $2\beta_2$, $2\beta_3$, etc., during which the suction valve is raised by the hook, are dependent upon the above-mentioned positions of the secant.

So far, merely the details of the distributing system have been considered;

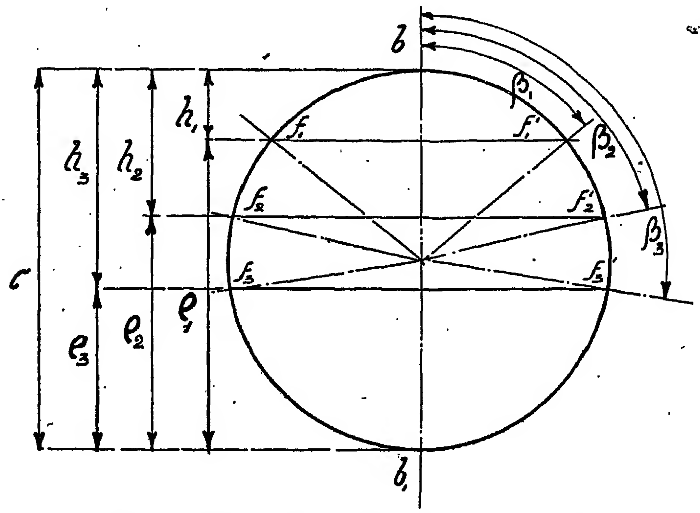


Fig. 291.—Positions of the Secants relative to those of the Governor Collar.

but of more interest is the action of this system, as will now be considered with reference to the pump plunger.

The action of the pump, if there were no distributing arrangement, might be represented (Fig. 292) by a circle of diameter equal to the stroke S of the plunger, in which the arc $a_1 m a$ would be the suction, and the arc $a n a_1$ the delivery stroke.

During the suction stroke the corresponding valve is automatically raised, and its lifts are to a certain scale, neglecting the obliquity of the connecting-rod, proportional to the heights d, which correspond to the various angles traversed by the crank.

Super-impose the two circles, which represent the diagrams of the plunger and of the distribution system, so that they have a common centre, and so that the axes of motion $a\,a_1$ and $b\,b_1$ contain the angle α, equal to that which the two eccentrics or the two cranks of the plunger and of the distribution system make with one another. The straight line $a\,a_1$ will cut the

circle representing the action of the distributing mechanism at $a'a_1'$. Strai[ght] lines are drawn from the centre of the diagram through f and f' to cut [the] circle representing the action of the plunger at f_1 and f_1' respectively.

The diagram (Fig. 293) will thus be complete: let α be the angle between the two cranks; for a given position of the governor corresponding to a maximum distance e between the extremities of the hook and of the suction valve, the latter will be raised by the hook through the arc $f'bf$, and will lift automatically, due to the effect of suction, through an arc $a_1'b_1f'a'$. And so altogether the valve will be raised for the whole time that the eccentric of the plunger takes to traverse the arc a_1af_1, and the effective delivery stroke will be reduced to the period f_1a_1—that is to say, s.

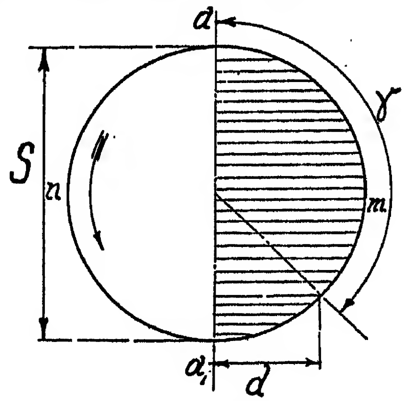


Fig. 292.—Diagram of Action of the Pump Plunger, shaded part representing suction stroke and plain part delivery stroke.

The governor, by displacing the straight line ff' parallel to itself, will correspondingly vary the distance e, and with it the useful stroke s.

Values of α for angles giving the desired regulation will now be considered.

Two conditions must be satisfied :—

(1) A part of the arc $f_1'af_1$, traversed by the eccentric of the plunger whilst the hook keeps the suction valve raised, must occur during the delivery stroke.

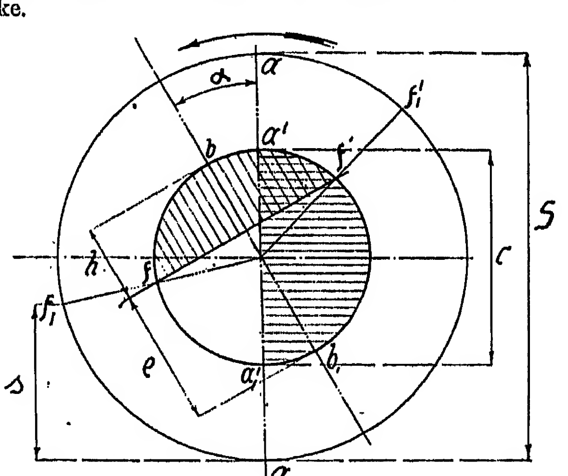


Fig. 293.—Combined Diagram of Actions of Plunger and Distributing Mechanism.

(2) Whatever may be the position of the governor between those which correspond to the delivery required for the engine at overload and for no delivery, the point f' at which the lift commences under the action of the

hook must be in the semi-circle $a_1'b_1a'$; or, in other terms, the lift must always commence during the suction stroke of the plunger.

The first condition is obvious and does not require explanation. The very great importance of the second is easily understood by considering that if the hook had to lift the valve during the delivery stroke of the plunger, when the whole pressure of delivery was upon the valve (45 to 75 atmospheres —640 to 1,065 lbs. per square inch)—the effort required would be considerable, and a large reaction would be transmitted to the regulating gear.

It is easy to be persuaded by sketching the corresponding diagrams that the angles α enclosed between 90° and 360° do not satisfy the aforesaid conditions, because those between 360° and 240° do not fulfil the second when the delivery is almost zero, those between 240° and 90° do not correspond for any position of the secant ff'—that is to say, for any position of the governor.

For $\alpha = 0°$ the action of the regulation takes place in a perfect way, for $\alpha < 90°$ the second condition is satisfied only if the maximum delivery required is less than that corresponding to the whole cylinder volume of the pump.

If, in fact, for action at maximum loads a delivery greater than that which corresponds to the useful stroke s_x (Fig. 294) were necessary—*i.e.*, the secant ff' had to be displaced as far as f_1f_1', so that the point f_1' would go further in the direction of motion than the dead centre a, the valve during the part $a f_1'$ would already be closed, and a useful delivery stroke would be commenced, to be interrupted by the action of the hook during a period $f_1'f_1$, and then recommenced when the point f_1 had been passed. The second condition would not then be satisfied.

The maximum useful stroke allowed to a given pump for a given value of α less than 90°, may then easily be found by considering the limiting condition in which the point f_1' is found on the axis $a\,a_1$. In this condition (Fig. 295)—

$$\sigma_x = c - k \sin \alpha,$$

but

$$s_x = \sigma_x \cdot \frac{S}{c},$$

from which

$$s_x = (c - k \sin \alpha) \cdot \frac{S}{c},$$

but

$$k = c \sin \alpha,$$

whence

$$s_x = (c - c \sin^2\alpha)\,\frac{S}{c},$$

$$s_x = c \cos^2\alpha \,.\, \frac{S}{c},$$

$$s_x = S \cos^2\alpha.$$

That is to say, if $\alpha < 90°$, *in order that the regulating gear may work correctly, the cylinder volume of the pump must be such that, with the engine at maximum load, the required delivery may be less than that corresponding to a useful stroke* S $\cos^2 \alpha$.

From this formula it results that the necessary cylinder volume increases rapidly with the value of the angle α to a limit when $\alpha = 90°$ and the useful

delivery stroke $s_x = 0$. *The practical values for this angle* α *are in fact* $\alpha =$ *from* $0°$ *to* $45°$ *inclusive.*

Further, to design the fuel injection pump and its regulating gear for a

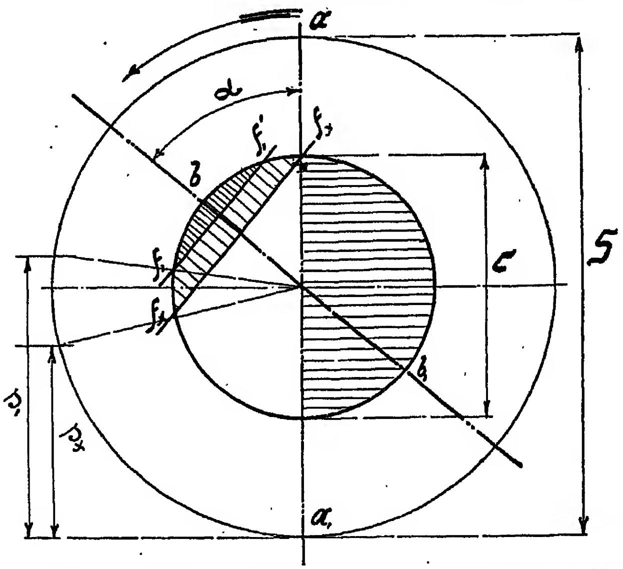


Fig. 294.—Combined Diagrams of Action of Plunger and Distributing Mechanism of a Pump unsuitable for a given Maximum Delivery.

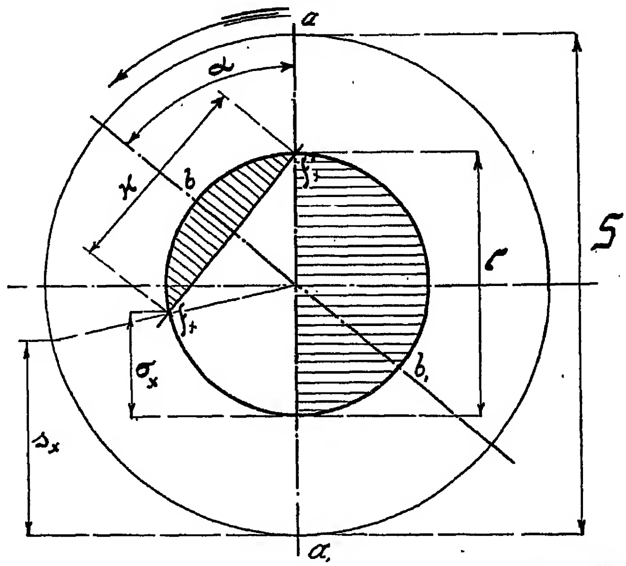


Fig. 295.—Combined Diagrams of Action of Plunger and Distributing Mechanism showing Maximum Useful Stroke of Plunger.

given engine, it is necessary, first to estimate closely the maximum hourly fuel consumption at full loads $F_{max.}$; to select a suitable value for α, and then to determine a cylinder volume for the pump, such that if A is the area of the cross-section and S the stroke of the plunger, the maximum estimated delivery will be

$$F_{max.} < 60\,n\,.\,A\,.\,S\cos^2\alpha,$$

in which n is the number of strokes per minute.

The value of the stroke c of the hook may then be determined by the aid of the diagram, after which it is possible to draw out the whole diagram of regulation (Fig. 296).

The useful stroke s_m corresponding to the engine at maximum load is immediately obtained from $F_{max.} = 60\,.\,n\,.\,A\,.\,s_m$.

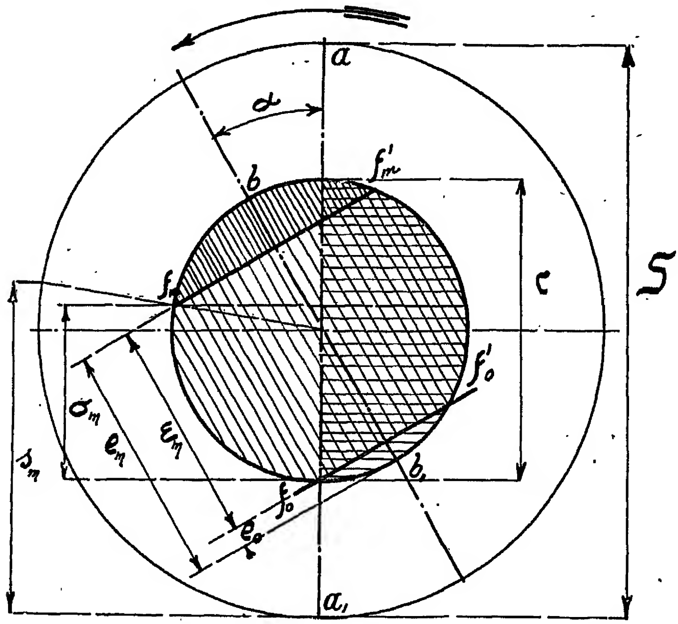


Fig. 296.—Combined Diagrams of Action of Plunger and Distributing Mechanism of a Pump suitably designed for delivery at all Loads from No-load to Maximum.

The secant $f_m f'_m$ can thus be drawn corresponding to the position of the governor when the engine is at full load. This secant, on being displaced parallel to itself for the varying powers of the engine, gives the various values of useful stroke s, and consequently the deliveries of the pump. When the secant reaches $f_o f'_o$ and passes through the bottom dead centre a_1, the useful stroke, and therefore the delivery, become zero.

Thus by varying the distance e between the extreme point of the hook and of the suction valve, from e_m to e_o—that is to say, displacing in space the travel of constant amplitude c of the hook by a distance $\varepsilon_m = e_m - e_o$, a variation of delivery from o to $F_{max.}$ is obtained.

In other terms, by assigning to the collar of the governor a stroke wh being transmitted by the levers to the link of the hook, will be transforn into the length ε_m, the complete regulation of the engine is achieved.

The corresponding values of the displacement ε, the useful stroke s and of $e = \varepsilon + e_0$ may be taken from the diagram (Fig. 297), or from the following equation :—

$$\varepsilon = a + b,$$

$$b = \frac{\sigma}{\cos \alpha},$$

$$a = (m - n) \sin \alpha = \sin \alpha \left(\sqrt{\sigma (c - \sigma)} - \sigma \tan \alpha\right),$$

whence

$$\varepsilon = \frac{\sigma}{\cos \alpha} + \sin \alpha \left(\sqrt{\sigma(c - \sigma)} - \sigma \tan \alpha\right) . \qquad (1)$$

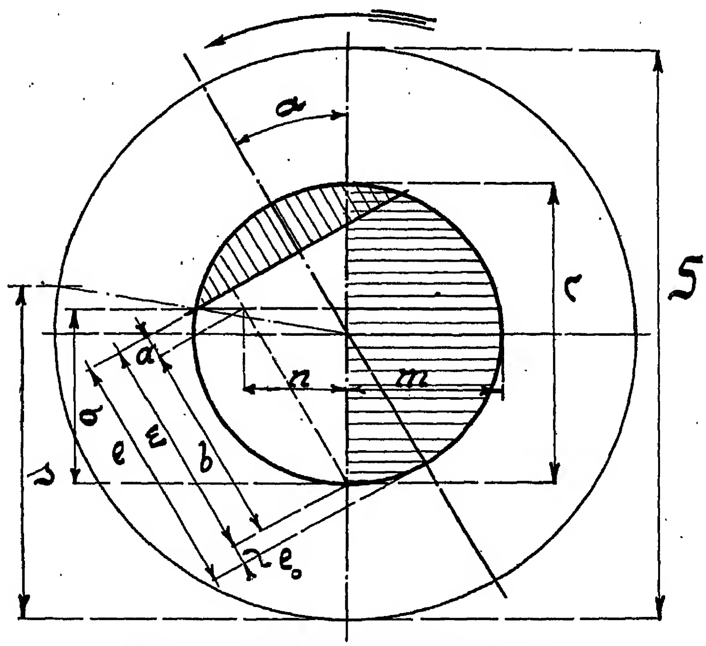


Fig. 297.—Diagram giving Values of ε, e, and s.

from which may be obtained σ and therefrom s with the aid of

$$s = \sigma \frac{S}{c}.$$

The values of the maximum distance e between the hook and the suction valve are to be found from

$$e = \varepsilon + e_o = \varepsilon + \frac{c}{2}(1 - \cos \alpha).$$

From the formulæ above it results that when $\alpha = 0$, $\varepsilon = \sigma$, and since $e_0 = 0$, therefore $\sigma = e$. That is to say, $s = e \frac{S}{c}$.

For every position of the governor, if $\alpha = 0$, the useful stroke of the pump is proportional to the maximum distance between the suction valve and the hook, in the ratio of the stroke of the pump plunger to the stroke of the hook. To apply the foregoing considerations to a particular type of pump for Diesel engines, a numerical example will be taken.

1. The calculation for a pump of the type shown in Fig. 274, p. 197, will be made, to design the regulation for a four-cycle Diesel engine of 30 normal B.H.P. and 36 maximum B.H.P. in each cylinder, with a speed of revolution of 220 per minute; the speed of the pump, which is driven from the horizontal cam shaft, will be 110 revolutions per minute. The cylinder volume of the pump must be such as to give a delivery equal to a consumption of 600 to 900 grammes (1·34 to 2·0 lbs.) of fuel per B.H.P. per hour at full output. A plunger diameter of 18 mm. (·71 inch), with a stroke also of 18 mm. (·71 inch), answers exactly to this condition.*

The total delivery resulting is—

$$18 \times \frac{\pi}{4} . 18^2 \times 110 \times 60 = 301{,}752 \text{ cub. mm. per hour,}$$

which is equivalent, supposing the specific gravity of the fuel to be 0·93, to a consumption of

$$30{\cdot}1752 \times 0{\cdot}93 = \text{about } 28 \text{ kg. per hour;}$$

i.e., $\frac{28}{36}$ = about 0·75 kg. of fuel per B.H.P. per hour at maximum load of 36 B.H.P.

It is known that an engine of the type under consideration, if well regulated, ought to consume at overload about 205 grammes (·452 lb.) per B.H.P. per hour of heavy oil with a calorific value 10,000 calories per kg. (18,000 B.Th.U. per lb.). At normal load the consumption should be 200 grammes (·441 lb.) per B.H.P. per hour, and at $\frac{3}{4}$ and $\frac{1}{2}$ load 210 grammes (·463 lb.) and 240 grammes (·529 lb.) respectively. It is easy, therefore, to calculate the delivery necessary for each load and by simple proportion, the corresponding useful strokes; †

$$s = \frac{F_s}{F_S} . S,$$

in which s = useful stroke of the pump required for supplying the engine at a given load.

S = total stroke of pump plunger.

F_s = delivery in grammes per hour corresponding to stroke s.

F_S = delivery in grammes per hour corresponding to stroke S.

* The plunger and its rod may have the same diameter provided this be suitable to withstand, without undue stress, the end on load calculated for a pressure of at least 80 kgs. per sq. cm. (1,140 lbs. per square inch).

† In marine engines in general, it is not necessary to trace the positions of the regulating mechanism corresponding to each intermediate load. The exception is the case in which the reversing system is external to the engine, and is such as to allow the engine to work at a constant speed (the Del Proposto and Föttinger systems). With these systems a centrifugal governor, identical with that for stationary engines, is substituted for the manœuvring wheel. If an engine is rigidly connected to the propeller, it is sufficient to design the regulating mechanism for full power and for the estimated minimum speed. To find this latter position, it will be necessary to calculate approximately the B.H.P. and the fuel consumption corresponding to that speed.

In the case under consideration, the values are as follows:—

Load.	Power in B.H.P. N.	Fuel Consumption in Grms. per B.H.P.-hour. K.	$F_s = KN$ gr.	$s = \frac{F_s}{28,000} \times 1{\cdot}8$ cm.
Maximum,	36	205	7,385	0·475
Normal,	30	200	6,000	0·386
Three-quarters,	22·5	210	4,725	0·304
One-half,	15	240	3,600	0·231

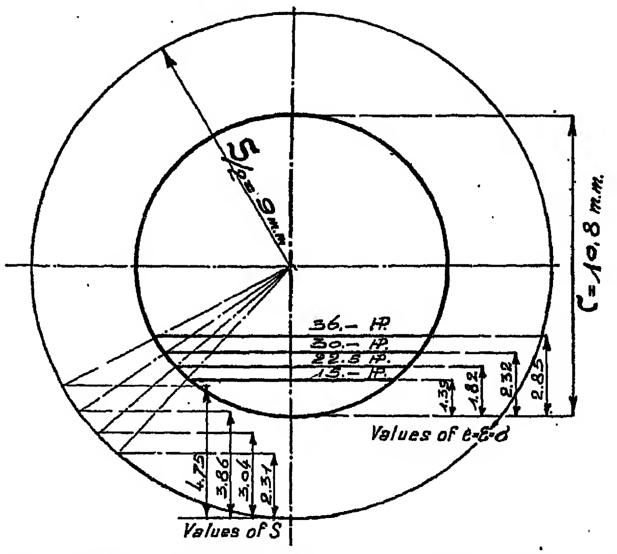


Fig. 298.—Diagram giving Values of e, ϵ, and σ, corresponding to values of useful strokes s when $a = 0$.

At this point a value must be assigned to the angle a. The calculation may be first made for a value $a = 0°$, and then for $a = 45°$. If $a = 0$, the equation (1) (p. 215) for the values of s and of ε becomes

$$\varepsilon = \sigma = s\frac{c}{S} = e.$$

Fixing the travel e of the hook at 10·8 mm. (·425 inch); then

$$e = \varepsilon = s\frac{10{\cdot}8}{18} = 0{\cdot}6\, s.$$

Substituting in this equation the values of s found above for the various loads (see Fig. 298)—*i.e.*, for B.H.P's. of 36, 30, 22·5, 15, the corresponding values of e in cm., 0·285, 0·232, 0·182, 0·139, are obtained. And so, if $a = 0$,

to obtain a delivery variable between the limits 0 and about 7·4 kgs. (16·35 lbs.) per hour, corresponding respectively to the point for stopping the engine and that for running at maximum power, it is sufficient if the travel of the collar of the governor be such as will vary the value of ε (which in this case is equal to the maximum distance e between the hook and the suction valve) between zero and 2·85 mm. (·112 inch).

Having selected a suitable governor for the engine with, say, a stroke of 20 mm. (·787 inch) to give the complete regulation, a system of levers must be designed to transmit the movement to the hook, giving it a certain additional movement below the zero point and above that for the delivery at maximum load.

This is done in order to take account of the volumetric efficiency of the pump, an efficiency which so far has not been considered. In the case under consideration, the levers may be designed so that, with a governor stroke of 20 mm., $\varepsilon = 3$ to 3·1 mm. (·118 to ·126 inch). Putting now $\alpha = 45°$, and leaving unchanged the other details of the pump, the values obtained for F_s and s for the various loads still hold good, but it is necessary to recalculate those for e and ε.

Before proceeding, however, it must be ascertained whether, for $\alpha = 45°$, the travel of 18 mm. is such that at maximum load the point f' is before the dead centre a'. That this may be the case, the following conditions must be satisfied (see p. 212) :—

$$s_m < \mathrm{S}\cos^2\alpha.$$
$$s_m < 18 \times \cos^2 45°.$$
$$s_m < 9 \text{ mm. } (0·354 \text{ inch}).$$

Since it has been found (see table on p. 217) that the useful stroke s corresponding to the maximum load is 4·75 mm. (0·191 inch), these conditions are fully satisfied, and the total travel of the pump plunger fixed at 18 mm. is sufficient.

The values of the displacements ε of the travel of the hook corresponding to the various loads are found from—

$$\varepsilon = \frac{\sigma}{\cos\alpha} + \sin\alpha\,\{\sqrt{\sigma(c-\sigma)} - \sigma\tan\alpha\},$$

$$\sigma = s\frac{c}{\mathrm{S}},$$

in which $\alpha = 45°$, $c = 10·8$ mm., $\mathrm{S} = 18$ mm., and s has the value found above—

$$\sigma = 0·6\,s,$$

$$\varepsilon = \frac{\sigma}{0·707} + 0·707\,(\sqrt{\sigma(10·8-\sigma)} - \sigma).$$

From the values of ε those of the maximum distance e between the hook and the valve will be easily deduced from

$$e = \varepsilon + e_o = \varepsilon + \frac{c}{2}(1-\cos\alpha) = \varepsilon + 5·4\,(1-\cos 45°)$$
$$= \varepsilon + (5·4 \times 0·293) = \varepsilon + 1·582 \text{ mm. } (0·0622 \text{ inch}).$$

N = B.H.P.	s mm.	σ mm.	ε mm.	e mm.
36	4·75	2·85	5·38	6·96
30	3·86	2·32	4·77	6·35
22·5	3·04	1·82	4·15	5·73
15	2·31	1·39	3·54	5·12

Thus, to obtain the entire regulation of the engine when $\alpha = 45°$, the displacement ε of the hook should be 5·38 mm. (0·212 inch) (Fig. 299). That is to say, a travel must be assigned to the collar of the governor such as to produce a displacement ε of 5·5 to 5·6 mm. (0·216 to 0·220 inch), to take into account as usual the volumetric efficiency of the pump.

2. For pumps of the type represented in Fig. 280 (p. 201), the calcula-

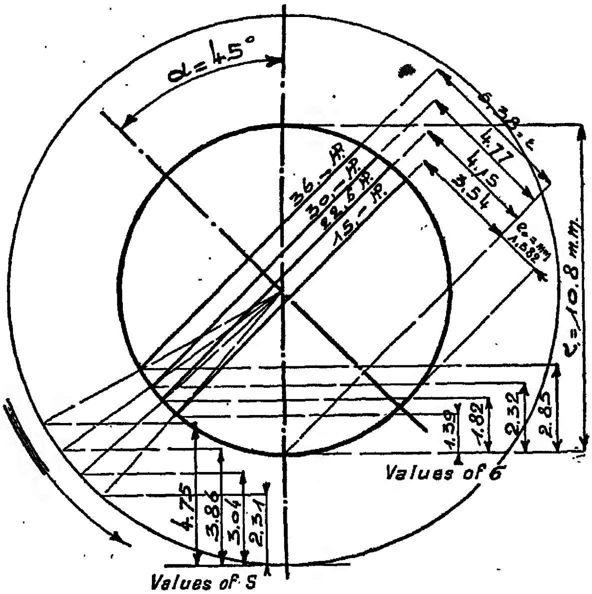


Fig. 299.—Diagram giving Values of e, ϵ, and σ, corresponding to values of useful strokes s when $\alpha = 45°$.

tion is the same; only, since the suction valve is inverted, all the considerations made for the values of α must be considered valid for an angle $\alpha + 180°$.

The same holds good within practical limits for the design represented diagrammatically in Fig. 281 (p. 202), in which the distributing eccentric transmits its movement to the valve by means of the angle-piece d.

3. For the type of pump represented by Fig. 286 (p. 205), in which the action of the governor varies the angle α according to the load, the calculation is made thus :—

The secant ff' is displaced in this case, not parallel to itself, but always tangential to a circle concentric with that of the diagram.

For this reason the value of e (Fig. 300) remains constant, but the values of ε and e_0, which modify the useful travel s, vary. To ensure that the two fundamental conditions of satisfactory regulation are fulfilled, the following must be true :—

$$e < \frac{c}{2}.$$

The delivery of the pump becomes zero when the position of ff' has reached $f_0 f_0'$—that is, when the point f coincides with a_1. The maximum delivery compatible with the two fundamental conditions is obtained when the point f' of the secant moves to f'_m and coincides with the same dead centre a_1.

The positions for maximum and minimum delivery are, therefore, symmetrical with respect to the axis of movement $a\,a_1$.

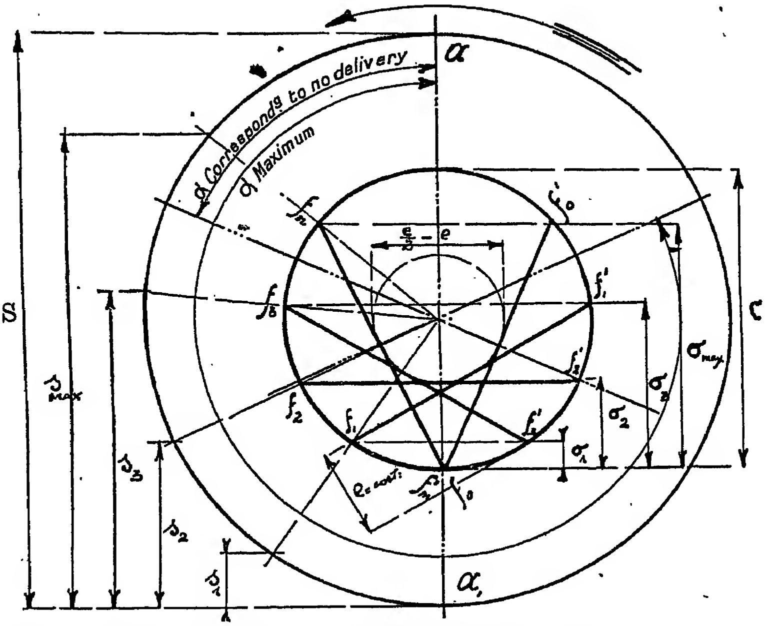


Fig. 300.—Diagram of Regulation of Pump in which the angle between Eccentrics working the Plunger and the Hook is varied by the Governor.

Fuel Injection Pumps and Regulation of Marine Engines.—The fuel pumps of marine and of stationary Diesel engines are almost identical, except that the control of the fuel delivered at the various loads, instead of being automatically effected by a governor, is carried out by means of a hand-actuated control wheel or lever.

In the case of marine engines, of course, the speed is not constant, and a reduction of power is always accompanied by a decrease in the rate of revolution, so that the centrifugal governor cannot be satisfactorily applied.

When the orders—*full speed, half-speed, and slow*—are given on the engine-room telegraph, the engineer moves the control lever or wheel, changing the

initial position of the hook under the suction valve of the pump, and thus varying the delivery of the latter to obtain the desired speed of revolution.

Greater variety of design is found with marine than with stationary engines in all their details, including the fuel injection pumps, in the case of which different methods of grouping and control are adopted, although the general action is usually the same.

The effects of the reactions on the governor gear are less to be feared, and may even be neglected by the designer, and in some cases indeed the usual methods of regulating, as fully described heretofore, may be abandoned, and for simplification a variable stroke plunger adopted, as is done in the Sabathé engine (Fig. 301).

In certain engines for cargo boats, it is preferred to have one single, large, and substantial pump for the whole engine, instead of providing one pump for each cylinder. Generally, however, in such cases, a second complete pump is fitted as a stand-by (Werkspoor).

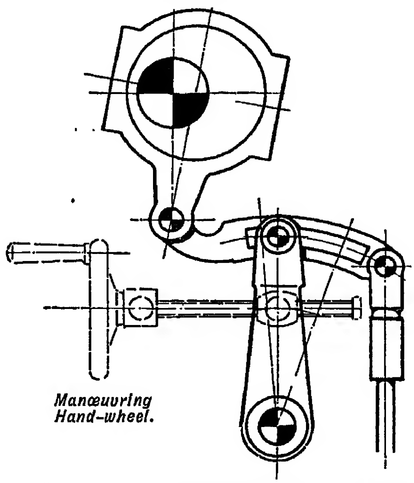


Fig. 301.—Method of obtaining Variable Stroke of Pump Plunger (Sabathé).

Safety Governors.—By means of the control lever, it is possible, as has been seen, to vary at will the power developed by the engine, and, consequently, the speed of the ship. It may happen that the resistance against which the engine is working diminishes so unexpectedly and suddenly that there is not time to control the engine before an excessive and even dangerous speed of revolution is attained.

In addition to the exceptional cases of breakage of the propeller shaft or of breakage or loss of the propeller itself, a great reduction or even almost a complete removal of the load often takes place in a heavy sea. In fact, due to the pitching of the ship, the propeller may partially or even totally emerge from the water, suddenly taking the load off the engine and allowing it to race.

In order that the engine should never run above a maximum given speed (at least with engines of any considerable power) a safety governor is provided.

This may consist of an eccentrically revolving mass held in by a centripetal spring. When the speed of revolution assumed by the engine increases above the normal by a certain amount (which may be 10 to 15 per cent.), the mass is thrown out so far from its axis by centrifugal force that in revolving it engages with and frees a trigger. A spring then pulls over a lever and immediately raises all the suction valves of the fuel pump, putting it suddenly out of action.

With such quickly acting governors it is necessary to replace the lever and the trigger in order that the engine may recommence firing. During the time employed in carrying out this operation the engine may stop altogether. Besides being inconvenient (necessitating a restart with compressed air), this might, under certain conditions, be dangerous. On this account a complete centrifugal governor similar to those fitted on stationary engines, but isochronous, is often preferred.* When the collar of the governor is in the highest position it holds up the suction valves of the pump; directly the speed is reduced below the determined maximum the collar returns automatically to its lowest position, and leaves the valves free to act normally.

In cases where the governor merely "throttles" the engine when it tends to race, due to emersion of the propeller, the Dunlop type of governor, which acts under the influence of the varying pressure of the column of water above a given point of the stern of the ship, may be applied.

These pressure governors have a quicker action than others, because they act even before the engine revolutions have accelerated, but are useless in cases of breakage of the shafting or of the propeller.

* A governor is isochronous when the centrifugal forces acting on the weights and the centripetal force of the spring are equal to one another for any position of the weights at a certain rate of revolution. If the speed is the least degree higher or lower than the number of revolutions per minute at which the governor is in equilibrium the collar flies to its corresponding extreme position.

CHAPTER XIII.

COMPRESSORS, AIR RESERVOIRS, AND SCAVENGING PUMPS.

THE high pressure air necessary for the injection of the fuel oil into the working cylinders during the running of the engine, and that which is used for starting the engine, is pumped by one or more compressors into cylindrical steel reservoirs. One of these—the smallest—is used for fuel injection air, and in the others, which are two in number with stationary plants, and several with marine installations, the air delivered in excess by the compressor is stored for starting.

The pressure necessary for fuel injection varies with the load and with the type of the engine, but is seldom lower than 45 atmospheres (640 lbs. per square inch) nor higher than 75 atmospheres (1,065 lbs. per square inch).

Starting, on the other hand, can be carried out with air of an initial pressure of 35 to 40 atmospheres (500 to 570 lbs. per square inch), and the air stored in one of the larger reservoirs should always be maintained at this or a higher pressure. The other, or others, in order to form a reserve of power, are charged to about 70 atmospheres (1,000 lbs. per square inch), to serve, in case of a failure to start, to bring the pressure in the first up to the value necessary. A system of valves and pipes connecting the reservoirs makes all these arrangements possible.

Compressors.—On account of the high pressure to be reached, the construction of compressors is not without its difficulties.

The compression is always arranged to take place in two or three stages, between each of which the air is cooled by passing through a reservoir surrounded by water. In some of the first engines, a two-stage compressor, properly speaking, was not used, as air was taken from the engine cylinder during the compression stroke at about 10 atmospheres (142·2 lbs. per square inch), and passed to a small high-pressure compressor. A mechanically operated valve similar to the starting valve was placed symmetrically with this latter in the cylinder head, and put the main engine cylinder into communication with the suction side of the compressor at a suitable period of the compression stroke. By this means the compressor was of small dimensions, but had few of the advantages of compression in more than one stage.

In engines of one cylinder only, one compressor is used, whilst those of more cylinders may have one or more compressors, sometimes, in fact, one for each cylinder. Multiplication of compressors is costly, although it certainly guarantees the production of a sufficient quantity of air, even when one of the compressors is out of working order. Where there are more than one compressor, the dimensions of each are reduced, and their construction is rendered more simple.

The methods adopted for driving and the positions selected for the compressor are many; the first—at the back of the crank case parallel to the

cylinders (Fig. 87)—is still the most common, and permits of the compressor being driven from the connecting-rod by levers. This disposition economises space and renders the compressor valves quite accessible, although the burnt oil which falls from the piston is bad for the link bearings, which are rather inaccessible owing to their position. When the compressors are so placed, multi-cylinder engines have generally one for each cylinder (Sulzer, M.A.N., Carels, etc.), but sometimes one or two for the engine (A.B.D.M. Stockholm, Nobel, etc.).

A different method of drive is that shown in Fig. 302 (G.M.A.) and Plate XII. (L.W.), showing a vertical compressor as generally used with high-speed enclosed engines. With slow-speed engines having independent framing for each cylinder, the compressor has its own frames similar to those

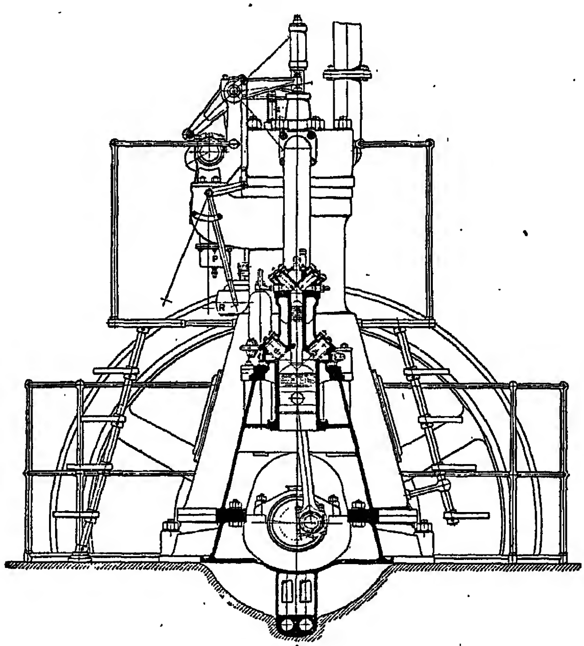


Fig. 302.—Vertical Compressor direct driven from Crank Shaft (G.M.A.).

of the main cylinders (Grazer, Güldner, S.L.M. Winterthur). This last arrangement is often adopted with four-cycle marine engines.

In large M.A.N. Augsburg vertical and horizontal engines the compressors are disposed as shown in Fig. 303.

The firm of Tosi often adopts with multi-cylinder engines a single compressor disposed vertically inverted in a suitable recess in the seating (Plate XI.)

Plate IX. (facing p. 192) shows a frequently used method of fitting the compressors horizontally. They are bolted to the end of the bed plate and driven by the crank shaft. With this arrangement engines of one cylinder have only one compressor, and those of more cylinders two (Langen & Wolf, G. F. Deutz, Benz, etc.).

Some constructors of note, instead of driving the compressor from the main engine, adopt a compressor separately driven by a belt or by an electric motor (Figs. 304 and 305, Grazer), which arrangement, especially with

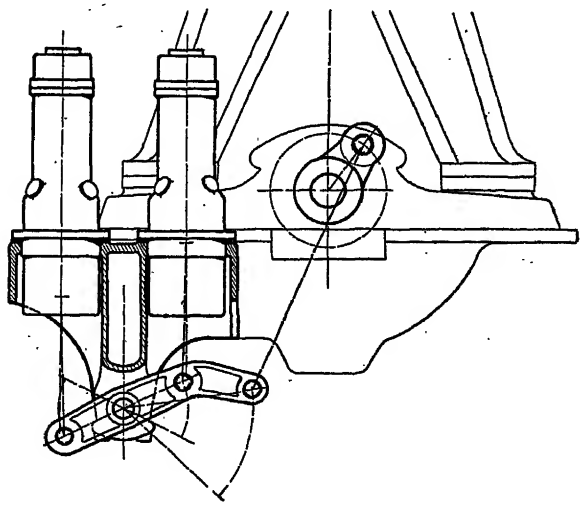


Fig. 303.—Air Compressors driven through a Balance Lever (M.A.N.).

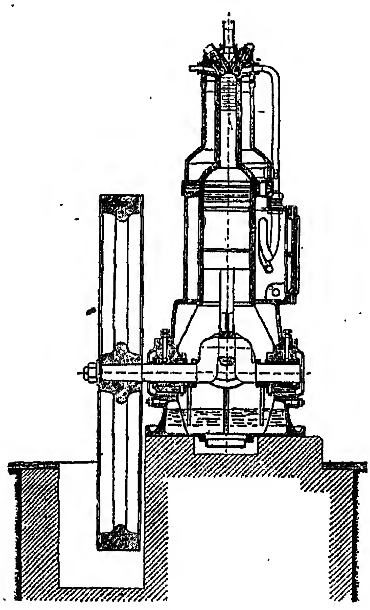


Fig. 304.

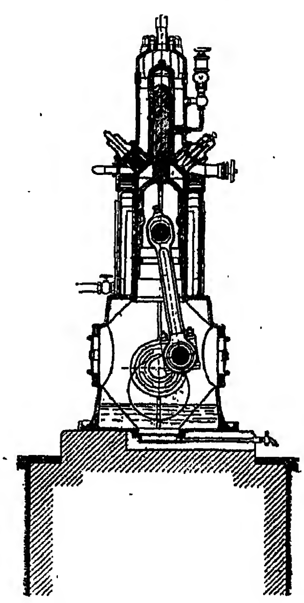


Fig. 305.

Figs. 304 and 305.—Air Compressor, belt driven by Electric Motor (Grazer).

the electrical drive, and if a battery of accumulators be installed to act as a stand-by for the electric generator, greatly minimises the risk of break-down.

As already explained, the maximum output of the compressor should be sufficient to charge the starting air reservoirs rapidly while furnishing the air for fuel injection. When the reservoirs are charged, the output of the compressor may be reduced to that necessary for the injection air at a pressure determined by the actual load on the engine. To accomplish this, arrangements are made to reduce the suction of the L.P. cylinder by means of a cock, a butterfly valve or a stop, which may be regulated to limit the lift of the suction valve.*

---

* In large two-cycle Sulzer engines, supplied as reserve plant for electrical power stations and subject in consequence to their variable load, an automatic arrangement fulfils the duty, usually left to the care of the watch-keeper, of regulating the output of the compressor and the injection air pressure in conformity with the load on the engine.

The lever of the governor (Fig. 306), besides acting on the fuel injection pump, is

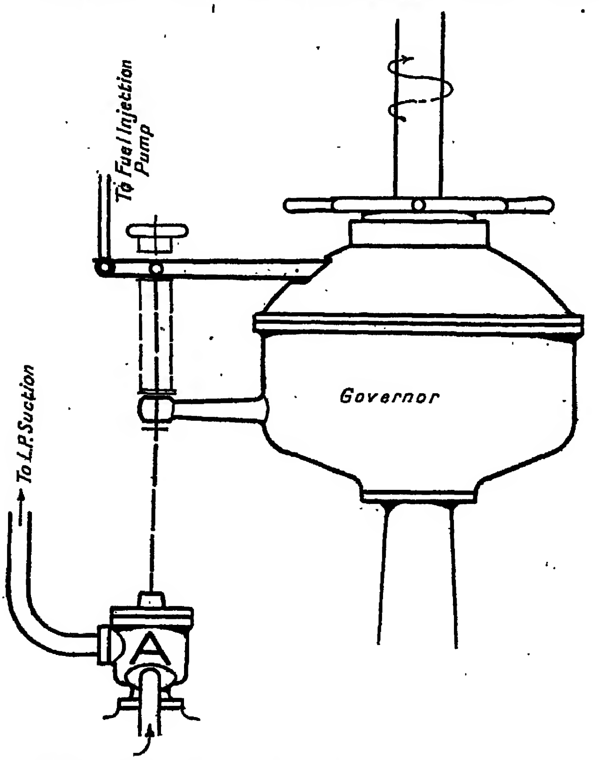


Fig. 306.—Diagram of Arrangement for automatically regulating Fuel Injection Air Pressure by Governor according to Engine Load.

connected to a double-seated valve A in the suction pipe of the L.P. cylinder of the compressor, to throttle the opening more or less, according as the governor is in the position corresponding to a small or large load. On account of the small dimensions of the fuel injection air reservoir, the output of the compressor has a rapid influence on the reservoir pressure and, therefore, on that of the fuel injection air.

A small wheel provided with a spring serves to set the valve A.

It is especially necessary that the clearance volume of the L.P. cylinder should be a minimum, and to accomplish this the bottom of the piston corresponds in shape as much as possible to that of the cylinder, and

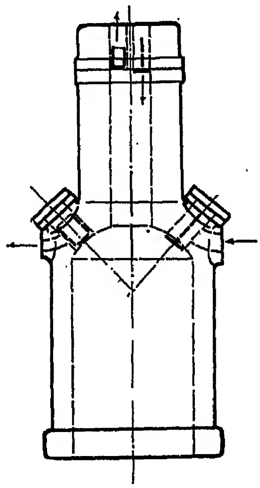


Fig. 307 (M.A.N. System).

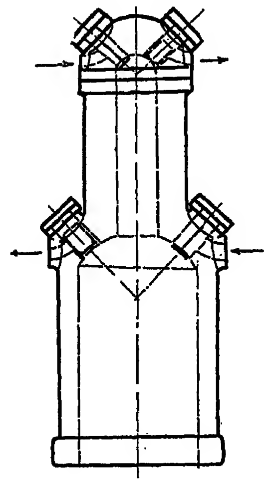


Fig. 308 (Langen & Wolf System).

Figs. 307 and 308.—Disposition of Valves of Air Compressors.

at the end of the stroke the surfaces almost touch. When the compressor is vertical, the valves are most frequently disposed as shown in Fig. 307. The bottom of the L.P. cylinder and that of its piston are dome-shaped,

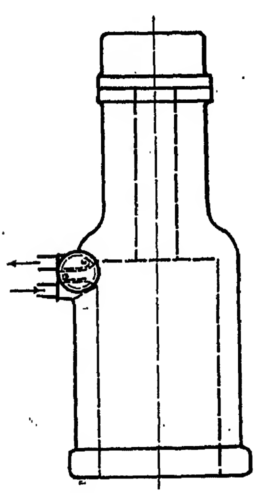


Fig. 309 (Sulzer System).

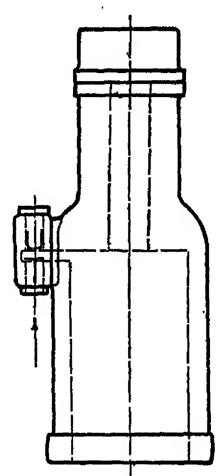


Fig. 310 (Sulzer and Tosi System).

Figs. 309 and 310.—Disposition of Valves of Air Compressors.

and the valves with their casings are disposed radially. The H.P. valves have their axes parallel, and are bedded in the flat H.P. head (M.A.N.). Sometimes, however, the H.P. cylinder has a spherical head with radial valves (Fig. 308, Langen & Wolf, M.A.N.).

For L.P. cylinders Gutermuth type suction and delivery valves (Fig. 309), enclosed in a single casing, are sometimes adopted (Sulzer).

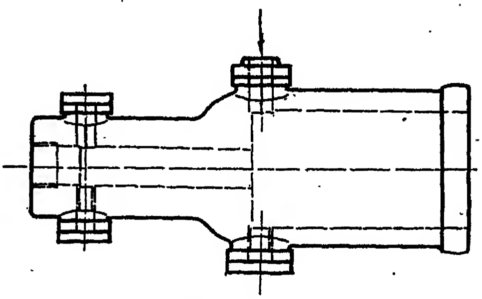


Fig. 311.—Disposition of Valves in Horizontal Air Compressor.

Fig. 310 shows the arrangement adopted by Sulzer and Tosi for the L.P. valves, enclosed in an external casing bolted to the compressor.

When the compressors are horizontal, the valves are sometimes arranged with vertical movement, as shown in Fig. 311 (Langen & Wolf, G. F. Deutz, Benz, etc.), or equally well as shown in Fig. 308, in which case their axes are at 45° to the horizontal.

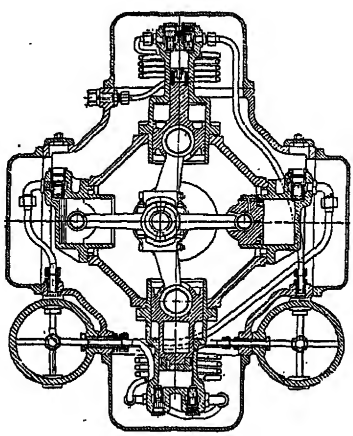


Fig. 312.

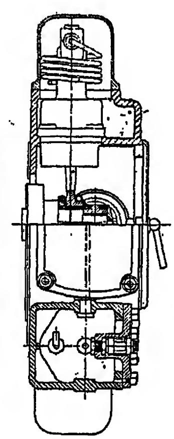


Fig. 313.

Figs. 312 and 313.—Reavell Air Compressor.

In some cases constructors, instead of making their compressors in their own works, buy them from firms who specialise in this type of machinery.

The compressors of Messrs. Reavell (Figs. 312 and 313), having four cylinders radially disposed around a single crank pin, form a good example

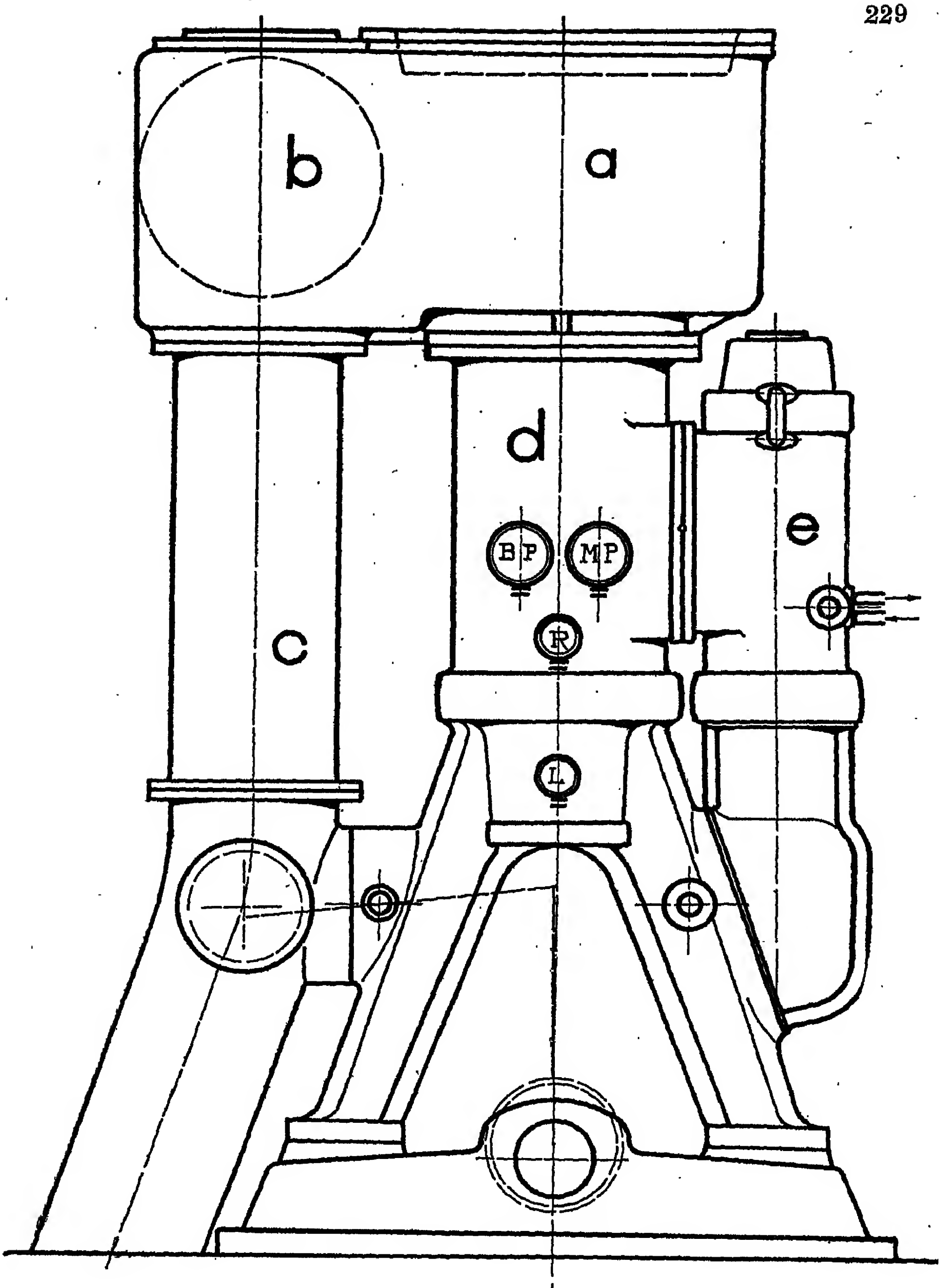


Fig. 314.—General Arrangement of Scavenging Pump and Air Compressors of Two-cycle Land Engine (Sulzer).

(Tosi, Sulzer, and some British constructors). The two L.P., the I.P., and the H.P. cylinders with their receivers are carried together in a single casing in which the cooling water circulates.

With large two-cycle engines, three-stage compressors are almost the rule, and sometimes the scavenging pump serves as the first stage, forming the L.P. cylinder of the compressor proper.

Fig. 314 shows the general arrangement of the scavenging pump and compressors of a two-cycle land engine. *a* is the scavenging pump with the cylindrical casing *b* for the distribution valves. The L.P. piston of the compressor working in the cylinder *d*, which is provided in its upper part with a group of Gutermuth flat valves, in this arrangement serves as a cross-head for the scavenging pump *a*. At *e* are the I.P. and H.P. cylinders of the compressor driven by a balance lever. Pressures in the three stages vary, as follows :—

L.P. = 3 kgs. per sq. cm. (42·66 lbs. per square inch),
I.P. = 17 to 20 kgs. per sq. cm. (240 to 285 lbs. per square inch),
and H.P. = 65 to 70 kgs. per sq. cm. (925 to 1,000 lbs. per square inch).

Fig. 315.

Fig. 316.

Figs. 315 and 316.—Views of Air Compressor End of Sulzer Two-cycle Land Engine.

The same arrangement of compressors and scavenging pump is shown in section in Plate VII. (facing p. 112), which represents those of the motor vessel "Monte Penedo."

Compressors are usually of cast iron, and are as often cast in one piece with their liners as with separate liners.

The valves, of bronze or of steel, are made with the greatest exactness, and are generally so designed that a cushion of air serves to render the return to their seating as gentle as possible.*

* To reduce valve dimensions, the valves are often duplicated, especially those of suction and delivery for the L.P. cylinder of large compressors (Fig. 303, p. 225).

The suction valves are generally of the poppet type; the delivery valves frequently cup-shaped. All, of course, are loaded with springs.

For the L.P. cylinders, besides the Gutermuth type of valves, ring or spiral valves of thin sheet steel are often used.

The pistons are of cast iron, and Fig. 317 shows a piston for a two-stage compressor. Tightness is assured by cast-iron rings, which, for the H.P. stage, are generally of such a small diameter that they cannot be sprung into place, but are carried in stepped carriers held in place by a junk ring fixed with a key or with a nut and lock nut.

The perfect machining of the piston and of the cylinder, and the selection of the material for, and the machining of, the rings, play an important part in the efficient working of a compressor.

The lubrication is of the utmost importance, and this should be continuous but not excessive. Too little oil might cause seizing or wear and leakage, whilst an excess of lubrication fouls the valves and impedes their

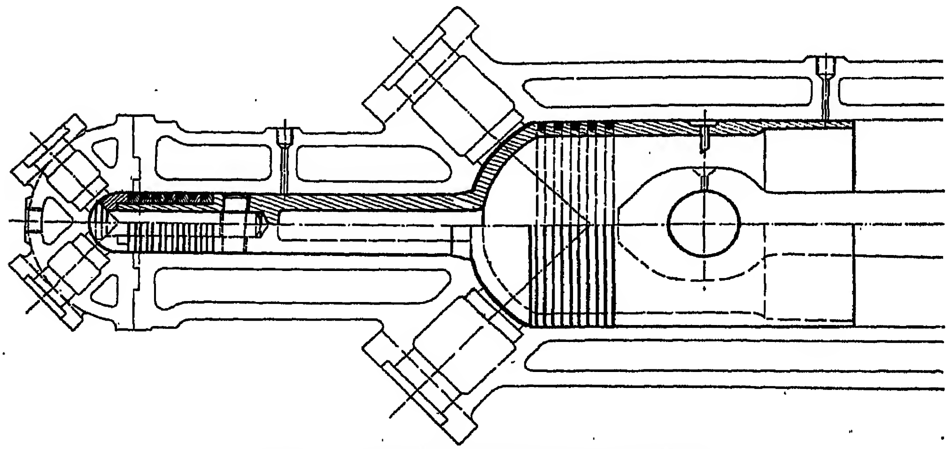


Fig. 317.—Piston for Two-stage Air Compressor.

ready working. In Fig. 317 are seen the oil leads, one for the H.P. and the other for the L.P. cylinders. Compressor lubrication cannot be carried out by a simple drip feed, because the inevitable leakage of air past the piston rings gives rise to pressure, interfering with the ready flow of the oil. One of the main engine forced lubrication pumps is used for the L.P. lubrication, whilst the H.P. requires a high-pressure lubricator.

It is good practice to fit a safety valve on the delivery pipes, to reduce the chances of bursting should the pressure rise to an excessive amount due to any error on the part of the engineer, or as a result of the obstruction of the passages.

Air Coolers.—The air which passes from one stage of compression to the other and from the H.P. cylinder to the reservoir should be cooled, and the connecting pipes must be of dimensions sufficient to form a receiver.

In some arrangements this receiver is cast with the compressor body and is cooled by the same cooling water. The cooling of the H.P. delivery

pipe is sometimes effected by passing it through the water lead to the compressor cylinder (Fig. 318, Sulzer).

Fig. 319 shows a unique type of cooler, in which the receiver and the delivery pipe of the H.P. stage are cooled (L. and W.).

The compressed air, on being delivered from the L.P. cylinder, after aving passed through a ribbed separator *a*, enters the receiver *b*, where it emains for cooling before passing to the suction side of the H.P. stage. Any water or lubricating oil in suspension in the air is arrested by the separator and collected at the bottom of the bottle-shaped receiver. During the running of the engine this receiver is blown down from time to time* by means of the small valve *e*. The H.P. part of the piping connected to the cooler is coiled round the receiver, and the coil and its flanges are well tinned.

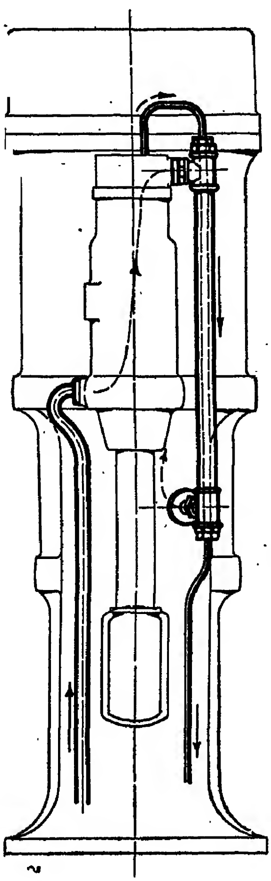


Fig. 318.—Sulzer Air Compressor Cooling Water System.

An accurate calculation is not required to determine the volume and the surface of this cooler, and the same type is satisfactory for engines of considerably varying powers. As an example, an apparatus as shown in Fig. 319, in which the receiver has a volume of about 9,000 c.c. (0·3177 cubic foot), and the coil a developed length of about 2·5 metres (8 feet $2\frac{1}{2}$ inches), with an external diameter of 30 mm. ($1\frac{3}{16}$ inches), may serve for engines of 50, 100, or even higher, B.H.P.

The volume of the receiver can be graphically determined by its influence upon the H.P. and L.P. diagrams, as with compound steam engines.

Calculations of Compressors.—It is not possible to determine by means of calculations alone the suitability of compressors to supply any given engine.

The maximum output of the compressor should be sufficient for the injection of the fuel when the engine is working at full load, and for charging the starting air bottles.

The time necessary to fulfil the second operation is in no way fixed and is not of primary importance. The calculation of the weight of air which passes from the fuel injection valve in a given time of opening, besides being exceedingly difficult, is rendered vague by the uncertainty in evaluating the loss of pressure and the coefficients of efflux of the air through the various orifices of the pulverising apparatus. Only practical experience can furnish the necessary information for determining compressor dimensions.

* With large units the separator has frequently a continuous automatic blow-down.

Since the types of compressors in use differ so little from one another (there are usually automatic valves, the smallest possible clearance

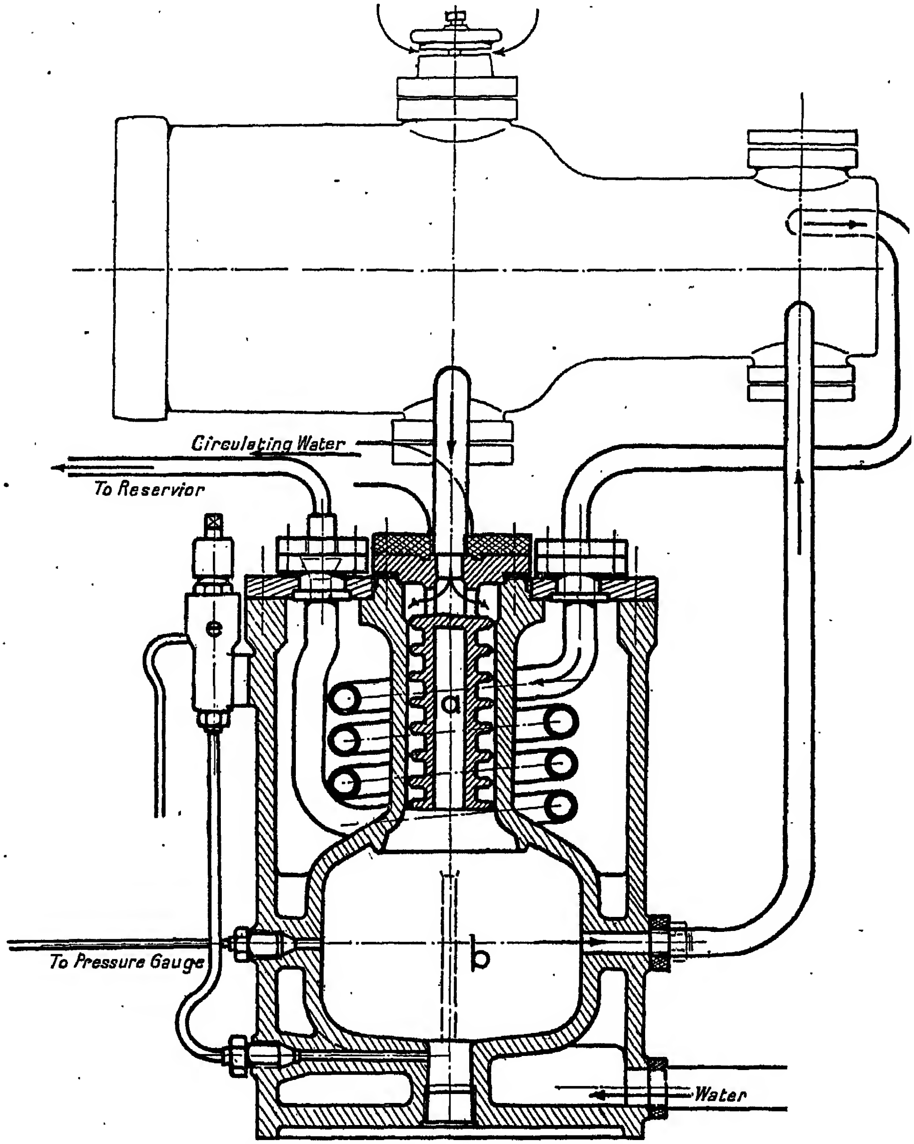


Fig. 319.—Air Compressor Intercooler (L. & W.).

spaces, comparatively low piston speeds, and the same number of revolutions as the main engine), it would seem to be sufficient to take the ratio between

the volume of the L.P. cylinder of the compressor and that of the main engine cylinders supplied by the compressor.

This ratio is not constant for all types of engines; for instance, in two-cycle engines the compressor must have a cylinder volume about double that necessary for supplying a four-cycle engine of equal cylinder volume, because the injection of the fuel takes place once every revolution instead of once every two revolutions. High-speed engines have larger compressors than those running at a low number of revolutions per min., since the holes of the pulveriser are larger (to permit of the rapid passage of the fuel), the pressure used is almost always higher and the volumetric efficiency of the compressor is lower. The capacity of the compressor also varies by a certain percentage in large as compared with small engines.

The L.P. cylinder volume of compressors for four-cycle engines varies generally between 5 per cent. (for large) and 7 per cent. (for small units) of that of the main engine cylinders of engines running at a normal speed of revolution, and from 9 to 11 per cent. with high-speed engines.*

With two-cycle slow-running engines of high power the L.P. cylinder volume of the compressor is about 10 per cent. that of the main engine cylinders, and 15 to 20 per cent. in the case of higher speed two-cycle engines. Once this rule is fixed, the volume of the L.P. cylinder can be obtained by fixing the piston speed between 0·50 and 1 metre (1·64 feet and 3·28 feet) per second for small compressors of engines of moderate speed, about 1·5 metres (4·92 feet) per second for medium-sized compressors, and 2 to 2·50 metres (6·55 to 8·2 feet) per second for the largest units. With high-speed engines the piston speed varies between 2·50 and 4 metres (8·2 and 13·12 feet) per second, according to the number of revolutions and the power.

The final pressure is reached in two or three successive compressions.

The multiplication of the stages of compression improves the volumetric efficiency of the compressor and diminishes the amount of work absorbed, besides allowing of better cooling of the air. The increase in the number of cylinders has its effect upon the price, so that small compressors have two stages, and the larger units three or even four stages with two-cycle engines in which the scavenging pump may serve for the first compression stage.

The minimum amount of work to compress a given weight to a given pressure is obtained when the compression is isothermal, and for that reason it is desirable to cool the cylinders by water circulation as efficiently as possible, in order to take from the cylinder walls the maximum possible quantity of heat. Even with water circulation, the cooling is not sufficient to reduce the mean index of the compression curve below 1·1, and the value usually met with in practice is between 1·5 and 1·3.

Having drawn the diagram of a single-cylinder compressor equivalent to the one it is desired to construct, the total area is divided into two or three parts, according as the compressor is of two or three stages.

Calculations are useful to assign values to the pressures of the various intermediate receivers. By deriving the expression for the work absorbed

* With the very smallest engines running at the highest revolution these figures may even be exceeded; for example, 14·5 per cent. with 5 B.H.P. 600 revolutions per minute engines made by R. Diesel & Co., Munich.

with regard to the pressure of the receiver p_2, the condition of minimum work is given by

$$p_2 = \sqrt{p_1},$$

where p_1 is the maximum pressure to be reached.

When compression takes place in three stages, if p_3 is the final pressure of the L.P. cylinder, that of the I.P. cylinder p_2 becomes—

$$p_3 = \sqrt[3]{p_1}, \text{ and } p_2 = (\sqrt[3]{p_1})^2.$$

Example.—To calculate the dimensions of a two-stage compressor for an engine of 60 to 70 B.H.P., at 220 revolutions per minute, with two cylinders of 270 mm. diameter, and a stroke of 410 mm. The delivery pressure of the compressor is to be 64 atmospheres.

The cylinder volume of the engine is

$$2 \times 23{\cdot}47 = 46{\cdot}94 \text{ litres.}$$

Allowing 6 per cent. of the main cylinder volume for that of the compressor L.P. cylinder, then—

The L.P. cylinder volume = 2·82 litres.

If the piston speed is 1 metre per second, the stroke of the pistons in tandem will be—

$$S = \frac{1{,}000 \times 30}{220} = 135 \text{ mm.}$$

With compressors of this type the clearance volume of both the stages can be arranged to be a little less than 2 per cent. of the cylinder volume, so that the total volume of the L.P. cylinder is about—

$$2{,}820 + 50 = 2{,}870 \text{ c.c. approx.}$$

At the end of the suction stroke this volume is full of air at a pressure of from 0·95 to 0·98 atmosphere. For simplification, the pressure may be assumed as that of the atmosphere, and the temperature of the air 15° C. Having fixed the final pressure of the first stage of compression as $\sqrt{64} = 8$ atmospheres, the volume occupied by this air can be found.

Given that the index of the compression curve is 1·2, then—

$$p_0 v_0^{1{\cdot}2} = p_2 v_2^{1{\cdot}2},$$

$$1 \times 2{,}870^{1{\cdot}2} = 8 \times v_2^{1{\cdot}2},$$

$$v_2^{1{\cdot}2} = \tfrac{1}{8} \times 14{,}000 = 1{,}750$$

and

$$v_2 = 530 \text{ c.c.}$$

This would be the volume to be assigned to the H.P. cylinder were it not for the existence of the clearance volume and the cooling of the air in the receiver.

Owing to the L.P. clearance volume, all of the air compressed in this stage does not pass through the L.P. delivery valve, and only

$$530 - 50 = 480 \text{ c.c.}$$

is delivered.

In addition, the cooling of the air in the receiver reduces the volume to be assigned to the H.P. cylinder to an extent which it is necessary to calculate.

$$\frac{273 + t_2}{273 + t_0} = \left(\frac{p_2}{p_0}\right)^{\frac{m-1}{m}},$$

and since $$m = 1{\cdot}2 \text{ and } t_0 = 15^\circ \text{ C.},$$

then $$\frac{273 + t_2}{288} = \left(\frac{8}{1}\right)^{\frac{0{\cdot}2}{1{\cdot}2}} = 8^{0{\cdot}167} = 1{\cdot}43,$$

and $$273 + t_2 = 410^\circ \text{ or } t_2 = 137^\circ \text{ C.}$$

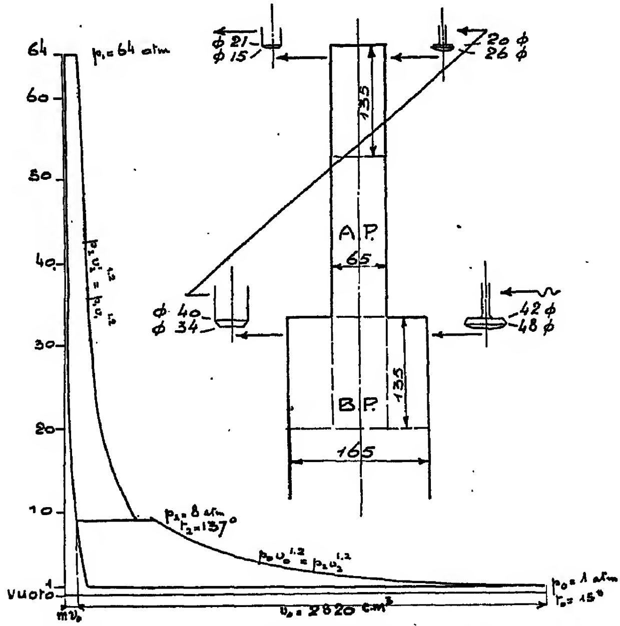


Fig. 320.—Diagram illustrating Calculation of Sizes of Compressor Cylinders.

Supposing that the temperature of delivery of the air from the receiver is 30° C., then the volume of air v'_2 becomes—

$$\frac{v'_2}{480} = \frac{273 + 30}{273 + 137} = \frac{303}{410} = 355 \text{ c.c. approx.}$$

This is the volume of air which should be drawn into the H.P. cylinder, but does not represent the H.P. cylinder volume, since the clearance space in this cylinder causes part of the piston stroke to be ineffective.

A certain quantity of air at 64 atmospheres pressure remains in the clearance volume at the finish of the inward stroke, and does not permit the suction of the next stroke to commence until the piston has moved outwards a

sufficient distance to allow the pressure in the H.P. cylinder to drop to a little less than 8 atmospheres.

Suction only commences after the piston has traversed a distance

$$s = \frac{m\,S\,(p_1 - p_2)}{p_2},$$

and since the clearance volume m is about 0·02 and the stroke 135 mm.,

$$s = \frac{0{\cdot}02 \times 13{\cdot}5\,(64 - 8)}{8} = \text{about } 2 \text{ cm.}$$

The useful stroke is reduced in this way to

$$S - s = 135 - 20 = 115 \text{ mm.}$$

To take into account the small drop of pressure below the 8 atmospheres necessary to give the air the required velocity through the valve, a useful stroke of 110 mm. may be taken, and the area of the H.P. piston will be

$$\frac{355}{11} = 32 \text{ sq. cm. approx.},$$

and the diameter d = about 65 mm.

The diameter D of the L.P. cylinder is obtained from

$$\text{c.c. } 2{,}820 = \frac{\pi}{4} S\,(D^2 - d^2),$$

$$\text{c.c. } 2{,}820 = \frac{\pi}{4} 13{\cdot}5\,(D^2 - 6{\cdot}5^2);$$

therefore, $D^2 = 271{\cdot}7$ and $D = 16{\cdot}5$ cm.

Recapitulating,

Cylinder diameter L.P.	= 165 mm.
,, ,, H.P.	= 65 mm.
Stroke	= 135 mm.

The valves are calculated for a given lift—always kept small—and for a given velocity of air. For a known drop of pressure the spring can be calculated for a desired air velocity.

For the example already taken, the valve would have the following dimensions :—

L.P. cylinder suction	diameter,	42 to 48 mm.
,, delivery	,,	34 to 40 mm.
H.P. cylinder suction	,,	20 to 26 mm.
,, delivery	,,	15 to 21 mm.

The following examples are those of compressors of well-known designs* :—

(1) Four-cycle engine of 5 B.H.P., $n = 600$, $D = 116$, $S = 150$. R. Diesel & Co., Munich. Compressor L.P. dia. = 70, H.P. dia. = 24 and S = 60.

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* In the examples given n = revs. per min., S = piston stroke. All dimensions are in millimetres.

(2) Four-cycle engine of 18 B.H.P., $n = 250$, $D = 215$, $S = 340$. M.A.N. Augsburg. Compressor L.P. dia. = 110, H.P. dia. = 30 and S = 100.

(3) Four-cycle engine of 25 B.H.P., $n = 220$, $D = 250$, $S = 400$. Langen & Wolf, Milan. Compressor L.P. dia. = 120, H.P. dia. = 34, S = 106.

(4) Four-cycle engine of 50 B.H.P., $n = 190$, $D = 340$, $S = 510$. Sulzer, Winterthur. Compressor L.P. dia. = 140, H.P. dia. = 40, S = 180.

(5) Four-cycle engine of 300 B.H.P., in three cylinders, $n = 175$, $D = 440$, $S = 640$. Langen & Wolff, Milan. Two compressors, L.P. dia. = 220, H.P. dia. = 65 and S = 270.

(6) Two-cycle four-cylinder marine engine 100 B.H.P., $n = 350$, $D = 180$, $S = 300$. M.A.N. Nürnberg. One compressor L.P. dia. = 190, H.P. dia. = 60, S = 110.

(7) Two-cycle marine eight-cylinder engine of 850 B.H.P., $n = 450$, $D = 300$, $S = 340$. M.A.N. Nürnberg. One compressor L.P. dia. = 350, H.P. dia. = 110, S = 260, or for the same engine two compressors, L.P. dia. = 260, H.P. dia. = 85 and S = 260.

(8) Two-cycle marine four-cylinder engine of 800 B.H.P., $n = 160$, $D = 470$, $S = 680$. Sulzer, Winterthur. One compressor L.P. dia. = 420, L.P. stroke = 600, I.P. dia. = 260, H.P. dia. = 100, and I.P. and H.P. stroke = 400.

(9) Two-cycle marine four-cylinder engine of 800 B.H.P., $n = 100$, $D = 460$, $S = 820$. Carels Frères, Ghent. One compressor (Reavell) L.P. dia. = 381 (2 L.P. cylinders), I.P. dia. = 241, H.P. dia. = 126·6, with a common stroke of 203.

Compressed Air Reservoirs.—The reservoirs are of steel, generally of a length of from 4 to 7 diameters to avoid excessive thickness and the consequent great weight. They are generally tested to about 80 atmospheres (1,137 lbs. per square inch), but can withstand very much higher pressures.

Fig. 321 shows diagrammatically in plan the arrangement of valves and piping which connect the three bottles together for land plants, and in Figs. 322 and 323 are seen the general arrangement of the group. The bottle I serves for fuel injection air, and II and III are those for starting air. Connected to I are the pipe which leads to the fuel injection valves, the compressor delivery pipe, and a third pipe leading to the branch piece of the starting air bottles. On these the valves a and a_1 (see Fig. 322) are fitted for supplying the starting air valves on the engines.

Supposing all the operations preparatory to starting the engines have been carried out, and that it is desired to start up, the valve b of the bottle I is opened after the pressure gauge has been put in communication with the bottle by means of the valve c. This gauge should show a pressure of about 50 atmospheres (710 lbs. per square inch).* If it is desired to start the engine from bottle III, for example, it is sufficient to open the valve a_1 and the engine starts on compressed air.† After several revolutions, when the engine

* When the motor starts, the compressor commences to work, and the valve b should always be open, otherwise the pressure might rise rapidly to a very high limit in the copper pipes between the compressor and the reservoir. No great reliance can be placed on the safety valves, as these are frequently rather crude and none too well designed.

† If it is desired to start with bottle III, the pressure in it should be about 35 to 40 atmospheres (500 to 570 lbs. per square inch), and should be verified by putting the gauge 2 in communication with the bottle by opening the valve f.

has picked up on fuel, the valve a_1 is closed, and only the valves *b* and *c* are left open. Before putting the load on the engine, the pressure in the bottle I should be raised to that suitable for the power desired, afterwards the valve *f* should be first opened and then the valve *d* gradually opened to recharge the bottle III to the pressure which it had prior to starting.

If the bottle II is the reserve bottle, it should be kept at a pressure of about 70 atmospheres (1,000 lbs. per square inch),* as may be verified on the gauge 2, by shutting *f* and *d* and opening *e*.

If the pressure has fallen, it is necessary, in order to bring it up to the original value, first to raise the pressure in I to about 70 atmospheres (1,000 lbs. per square inch), then having shut *f*, to open *d* and *e*. During this operation the pressure in the bottle I might be considerably higher than that suitable for the actual load at which the engine is running. The air at the fuel injection valve is then maintained at the desired pressure by throttling the

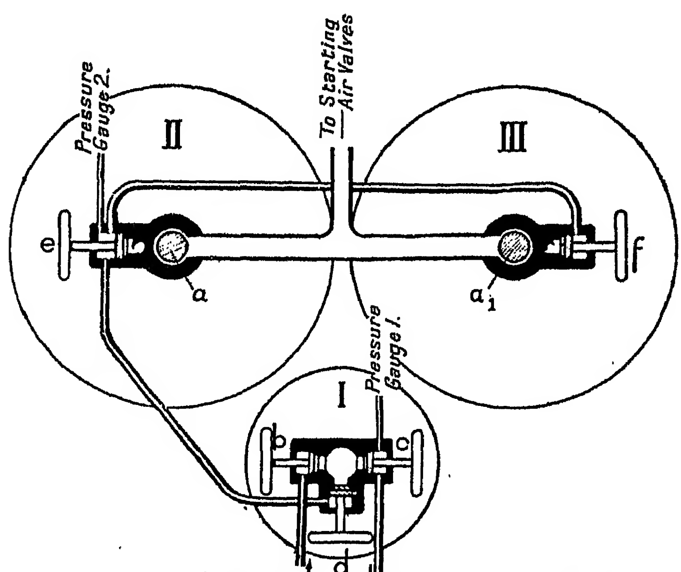


Fig. 321.—Plan Diagram of Valves and Connections for Compressed Air Reservoirs (Land Installation).

opening of the valve *c*. From Fig. 321 it will be seen that the points at which the gauges are connected are so arranged that during this operation the effective pressure of the fuel injection air can always be seen on the gauge 1, whilst the pressure in the reservoir I may be read from the gauge 2 if the valve *d* is open. When the charging of all the bottles has been completed by working the compressor at its full output, all the valves are closed except

* It is not prudent to increase this pressure, as with variations of the room temperature, especially in summer, the pressure in the closed bottle might reach a dangerous limit without being observed.

b and *c*, and the suction of the compressor is throttled until the output is exactly that required for the injection of the fuel at the required pressure.

The group of valves on the bottles do not constitute a very simple arrangement, but, in view of the number of manœuvres which can be carried out, they represent an ingenious solution of a rather difficult problem.

In addition to the gauges 1 and 2, there is generally a third in communication with the compressor receiver (B.P. in Fig. 322), which shows the pressure in the L.P. cylinder of the compressor. In Fig. 323 the small valves *p p* per-

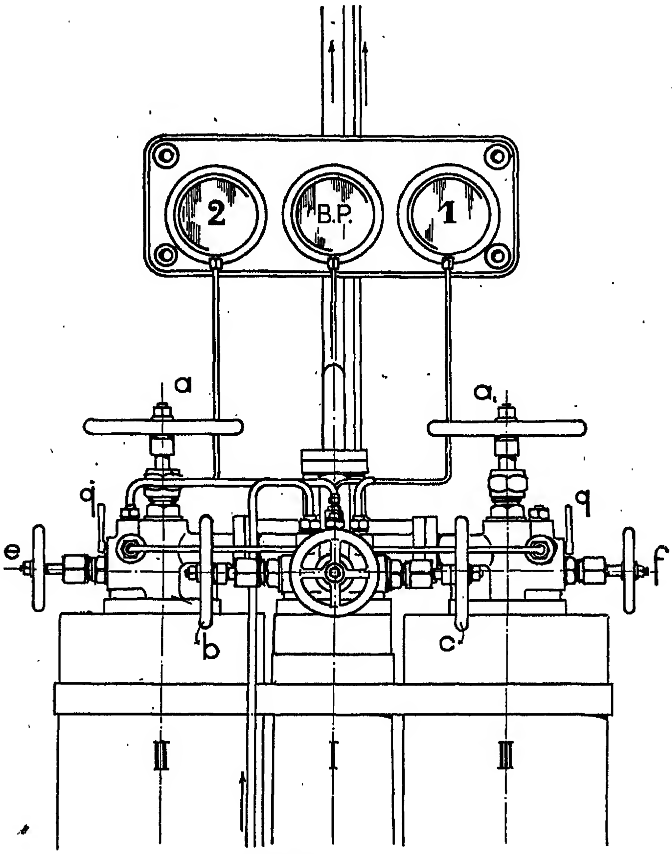


Fig. 322.—Front View of Starting and Fuel Injection Air Reservoirs and Connections (Land Installation).

mit of communication with the atmosphere by means of a drain pipe leading to the bottoms of the bottles. This serves from time to time for draining off any oil or water which may be carried in by the air. The same valves serve to discharge the reservoirs when it is necessary to grind in the valves or renew joints. The other small needle valves *q q* (Fig. 322) serve to

discharge the air from the gauge pipes, in order to bring the gauge pointer back to zero when the engine is not working.

The capacity of the reservoirs is not determined by any precise rule. Those for fuel injection should constitute the reserve for the compressor, and should have such dimensions as to make them act with reference to its delivery as does a flywheel to an engine. Those for starting should be of

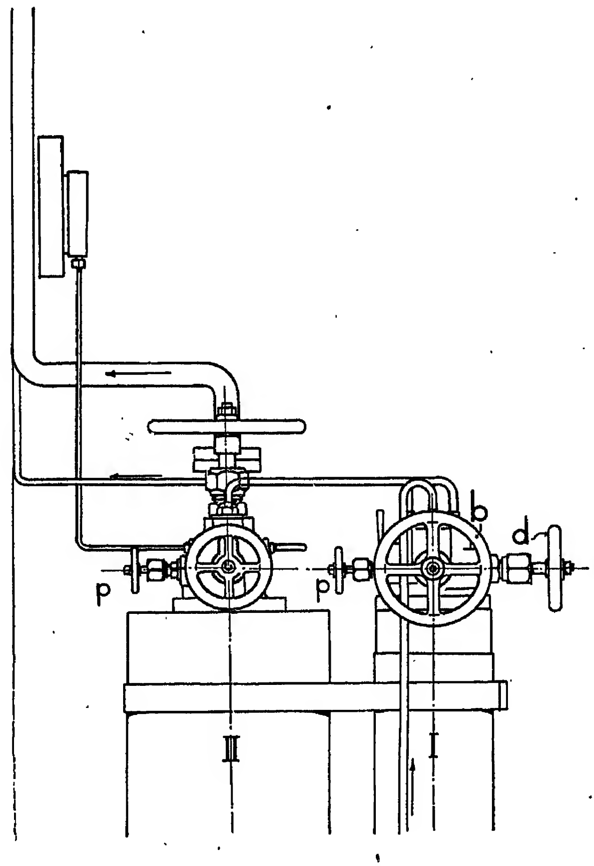


Fig. 323.—Side View of Starting and Fuel Injection Air Reservoirs and Connections (Land Installation).

sufficient capacity for five or six starts with land engines, and for a larger number in the case of marine plants.

It is understood that this indication is very vague, because starting may absorb different quantities of air according to the period of running on starting air.

It serves no useful purpose to run land engines on starting air for any appreciable time, for if the motor does not pick up immediately on fuel,

it were better to stop and to examine the more important parts for defects than uselessly to consume the air, which is always costly.

With marine engines this is not possible, since for the safety of the ship the engine must carry out immediately the orders given from the bridge. The order given on the engine-room telegraph must be absolutely obeyed, though it may necessitate running on compressed air at the cost of completely emptying the reservoirs. For this reason, the main compressor is always supplemented by an auxiliary of a capacity at least half and almost always about two-thirds that of the main compressor. This auxiliary compressor is always run during the periods when the main engine is manœuvring.

For land engines the capacity of the fuel injection air reservoirs varies from 0·4 to 0·6 litre (0·01412 to 0·02118 cub. ft.) per B.H.P., and those for starting air between 8 litres (0·282 cub. ft.) per B.H.P. for the smallest, to 3 litres (0·106 cub. ft.) per B.H.P. for medium-powered engines; for large engines the storage capacity is still lower.

With marine engines there are always more than two starting air bottles, forming a working as well as a reserve group, and sometimes with a third group forming an extra reserve. The connections and the valves between the various groups, bottles, and compressors, are naturally a little more complicated, but in principle are similar to those described for land plants.

The capacity of the starting air reservoirs varies with the type of work for which the boat is built, and with the personal ideas of the designer of the machinery. As an example, tugs which are required to be constantly manœuvring should have a relatively greater installation for producing and storing compressed air than large ships which are in the hands of tugs when manœuvring in and out of port.* Submarines and submersibles, in which propulsion under water is obtained with electric motors supplied from accumulators, and driving the main line of shafting, only require a small storage of starting air, since if the necessity should arise, the ship may be manœuvred on the surface by means of the electric motors. In fact, it is believed that in some Körting engines for German submersibles no provision of compressed air is made for starting the engines, reliance being placed entirely upon the electric motors for this duty.

The following are some examples of the capacity of air storage for marine installations :—

(1) Tug No. 21 of the Royal Italian Navy. Sulzer engine of 100 B.H.P., n† = 380. Four starting air bottles with a total capacity of 600 litres (21·18 cubic feet)—*i.e.*, 6 litres (0·2118 cubic foot) per B.H.P. Fuel injection bottle of 52 litres (1·835 cubic feet) or 0·52 litre (0·01835 cubic foot) per B.H.P.

(2) "Wotan"—herring fishing boat with Frerichs engines of 90 B.H.P., n = 300. Two starting air bottles with a total capacity of 520 litres (18·35 cubic feet) or 5·8 litres (0·205 cubic foot) per B.H.P. Fuel injection air bottle of 50 litres (1·765 cubic feet)—*i.e.*, about 0·56 litre (0·0198 cubic foot) per B.H.P.

* The tug, "Frerichs," is provided with engines of 200 B.H.P., which can reverse 60 times in succession by means of its compressed air storage, and its compressors are of such a capacity that thirty manœuvres may be carried out per hour without any diminution in pressure of the starting air.

† n = number of revs. per min.

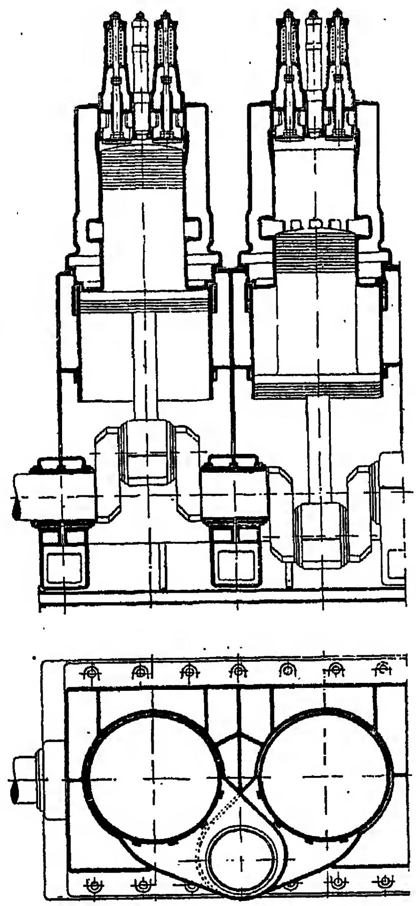
(3) "Vulcanus"—a tank ship with a Werkspoor engine of 500 B.H.P., $n = 180$. Four starting air bottles of 1·8 cubic metres (63·5 cubic feet), equal to 3·6 litres (0·1270 cubic foot) per B.H.P. Auxiliary compressor of 40 to 50 B.H.P.

(4) "Fordonian"—freight ship for the Canadian Lakes, with a Carels engine of 800 B.H.P., $n = 100$. Four starting air bottles with a total capacity of 2,680 litres (91·1 cubic feet), or 3·35 litres (0·118 cubic foot) per B.H.P. One fuel injection bottle of 66 litres (2·33 cubic feet), about 0·08 litre (0·0028 cubic foot) per B.H.P. An auxiliary compressor with a capacity of 3,200 litres (113 cubic feet) of free air per minute.

(5) "Rolandseck"—cargo boat with Carels-Tecklenborg engine of 1,500 B.H.P. at 120 revolutions per minute. Five starting air bottles with a total capacity of 4,000 litres (141·2 cubic feet) or about 2·7 litres (0·0954 cubic foot) per B.H.P., and an auxiliary compressor of 100 B.H.P.

(6) A 300 B.H.P. Sulzer submarine engine, running at 500 revolutions per minute, with starting air bottles of 200 litres (7·06 cubic feet) or 0·67 litre (0·02365 cubic foot) per B.H.P., and a fuel injection air bottle of 40 litres (1·412 cubic feet) or about 0·13 litre (0·00459 cubic foot) per B.H.P.

Scavenging Pumps.—The function of the scavenging pump, as already pointed out, is to compress air to a low pressure to free the working cylinders of two-cycle engines of the exhaust gas, and to charge them afresh with pure air for the next combustion stroke.



Figs. 324 and 325.—Scavenging Pumps (Fiat Type).

When the overall space is restricted, especially in the case of the lightest marine engines, each working cylinder has a stepped piston—*i.e.*, one having two diameters, of which the greater serves as the scavenging pump (Figs. 324, 325, Fiat, 326 and 327, Kind).

In this case there are as many scavenging pumps as working cylinders, and the pumps have the same stroke and piston speed as the working pistons. The valves for such pumps may be automatic (Fig. 327, Kind), or may be of the piston type (Fiat, Fig. 325, Körting, Fig. 255, p. 177).

It is often preferable to provide one or two double-acting scavenging pumps for the whole engine to make this part of the engine more complete; to obtain scavenging with colder and consequently denser air; to minimise

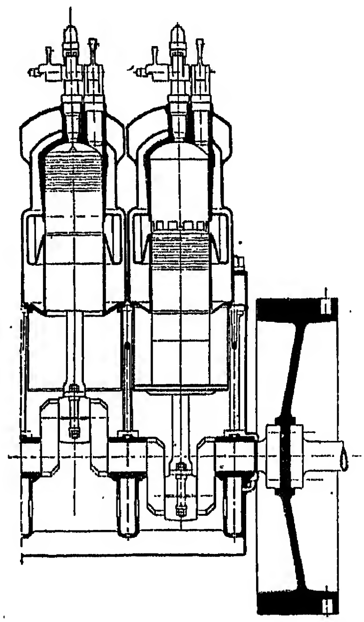


Fig. 326.

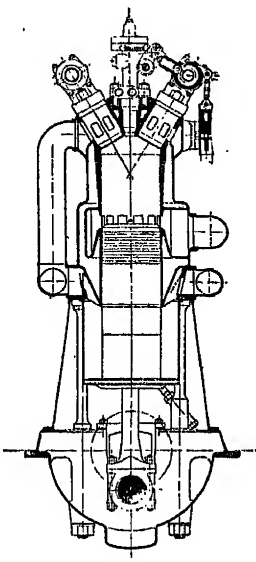


Fig. 327.

Figs. 326 and 327.—Stepped Piston Scavenging Pumps (Kind Type).

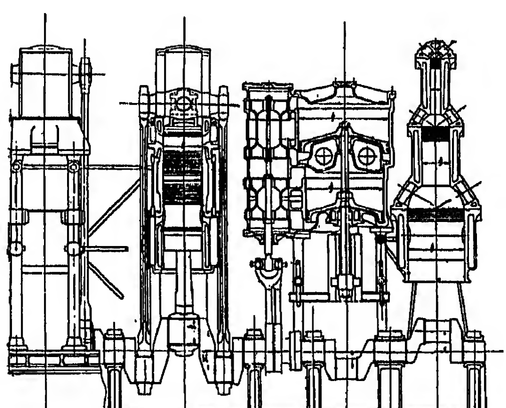


Fig. 328.—Junkers Submarine Engine with Scavenging Pumps having Piston Valves.

any increase of the reciprocating weights; and to keep the pumps away from the ill effects of the burnt lubricating oil, which falls from the pistons on to the scavenging pump plungers in the case of stepped-piston engines.

As a general rule small pumps have automatic valves, whilst those of large size are provided with piston valves (Plate VII., facing p. 112, and Fig. 328).

Automatic valves should always be as light as possible, and are sometimes of the Gutermuth type. These have the advantage that the clearance volumes are reduced to a minimum, and so the volumetric efficiency is thus augmented. This type does not give the same certainty and safety of action as do piston valves.

The cylinders of these pumps are of cast iron, and are not water-cooled; the pistons are of cast iron or cast steel made of as light section as possible; the valves also are generally of cast iron.

The general construction of this important part of the main engine is seen in the figures already cited, and in Fig. 328, representing a Junkers submarine engine. In this case there are two scavenging pumps placed in tandem with pistons of different diameters for reasons of accessibility in dismantling, and not, as might be thought, for two-stage compression.

Besides the method of driving these pumps by means of a separate crank shaft, the system of balance-levers, as shown in Plate XVII. (facing p. 272), is frequently adopted.

With reversible engines, the valves of the scavenging pump, if positively operated, are set for the new direction of rotation, as explained in Part II., Chapter XI.

The pressure of air in practice is generally between 0·15 and 0·25 kg. per sq. cm. (2·13 and 3·55 lbs. per square inch), and the maximum varies from 0·4 to 0·5 kg. per sq. cm. (5·7 to 7·1 lbs. per square inch).

The mean piston speed with stepped piston engines is, as already stated, the same as that of the main engine piston and with independent pumps usually varies from 2 to 3 metres (6·50 to 9·84 feet) per second.

The cylinder volume of the pump may be taken as from 1·6 to 1·8 times that of the cylinder served by the pump, if the latter has piston valves, and, therefore, has a relatively large clearance volume (0·12 to 0·18). The ratio may be reduced to 1·4 or 1·6 if the pump has automatic valves.

In the table on p. 246 some examples of the dimensions of scavenging pumps of modern engines are given.

EXAMPLES OF THE DIMENSIONS OF SCAVENGING PUMPS OF MODERN ENGINES.

Main Engine Particulars.								Scavenging Pump Particulars.							
B.H.P.	Type.	Maker.	No of Cylinders.	Revs. per Min.	$\frac{\text{Diameter}}{\text{Stroke}}$ of Cylinder in mm.	Volume of Cylinders served by One Pump. C. Litres.	Volume of Cylinders served by One Pump. Cub. Ft.	No.	Type.	Type of Valves.	$\frac{\text{Diameter}}{\text{Stroke}}$ of Pump in mm.	Piston Speed in Feet per Min.	Volume of One Pump. L. Litres.	Volume of One Pump. L. Ft.^{3}	Ratio. $\frac{L}{C}$
2,000	Land,	Sulzer,	4	150	$\frac{660}{800}$	548	19·4	2	Double-acting,	Piston valve,	$\frac{1,100}{500}$	492	950	33·5	1·74
1,500	Marine,	Carels,	6	120	$\frac{510}{920}$	564	19·95	2	Double-acting,	Piston valve,	$\frac{870}{760}$	590	900	31·8	1·60
1,000	Land,	Sulzer,	4	150	$\frac{500}{720}$	564	19·95	1	Double-acting,	Piston valve,	$\frac{1,100}{500}$	492	950	33·5	1·70
850	Marine,	M.A.N.,	8	450	$\frac{300}{340}$	24	·847	8	Single-acting,	Automatic valves,	$\frac{470-300}{340}$	1,000	35	1·235	1·45
800	Marine,	Carels,	4	100	$\frac{460}{820}$	272	9·60	2	Double-acting,	Piston valve,	$\frac{692}{586}$	394	448	15·8	1·65
750	Land,	Sulzer,	3	150	$\frac{500}{720}$	423	14·95	1	Double-acting,	Piston valve,	$\frac{1,000}{500}$	492	784	27·7	1·85
310	Marine,	Sulzer,	4	220	$\frac{310}{460}$	139	4·91	1	Double-acting,	Automatic valves,	$\frac{625}{350}$	505	215	7·6	1·55
100	Marine,	Sulzer,	4	380	$\frac{180}{250}$	25·4	·897	1	Double-acting,	Automatic valves,	$\frac{365}{200}$	500	42	1·48	1·65
100	Marine,	M.A.N.,	4	350	$\frac{180}{300}$	7·6	·2685	4	Single-acting,	Automatic valves,	$\frac{280-180}{300}$	690	10·8	·382	1·42
100	Marine,	M.A.N.,	4	600	$\frac{175}{200}$	4·8	·170	4	Single-acting,	Automatic valves,	$\frac{275-175}{200}$	790	7·0	·247	1·47

CHAPTER XIV.

ENGINE-ROOM, FOUNDATIONS, ACCESSORIES, PIPING OF LAND ENGINES.

Engine-Room.—A heavy oil engine may be installed in any engine-room, provided this be sufficiently well lighted and of a height to permit of piston withdrawal in the case of vertical engines. The atmosphere should be clean and free from dust.

For normal-speed vertical Diesel engines, the minimum height of the engine room necessary for withdrawing the piston (inclusive of the tackle, and the traveller on which this runs) is ten or eleven times the piston stroke. To assist the builder in drawing out his preliminary plans a formula has been deduced from data obtained from a large number of engines of different makes (Tosi, Sulzer, L.W., M.A.N., and G. F. Deutz), which, with close approximation, gives this minimum height in metres—

$$H = 3{\cdot}20 + 0{\cdot}04\,N,$$

in which N is the normal B.H.P. of the engine, or of one cylinder with multi-cylinder engines.

It is difficult to give any reliable rules for the ground plans of the engine-room, although generally it may be more or less roomy on a basis of the overall dimensions of the engine according to the available space. Formulæ for this have also been elaborated as obtained from the same engines, and are sufficiently approximate for average four-cycle engines.

The breadth in metres between the foot of the ladder leading to the starting platform and the flange on the opposite side of the bed plate is—

$$C = 1{\cdot}3 + 0{\cdot}02\,N,$$

in which N has the same signification as in the preceding formula.*

The maximum length of the engine is—

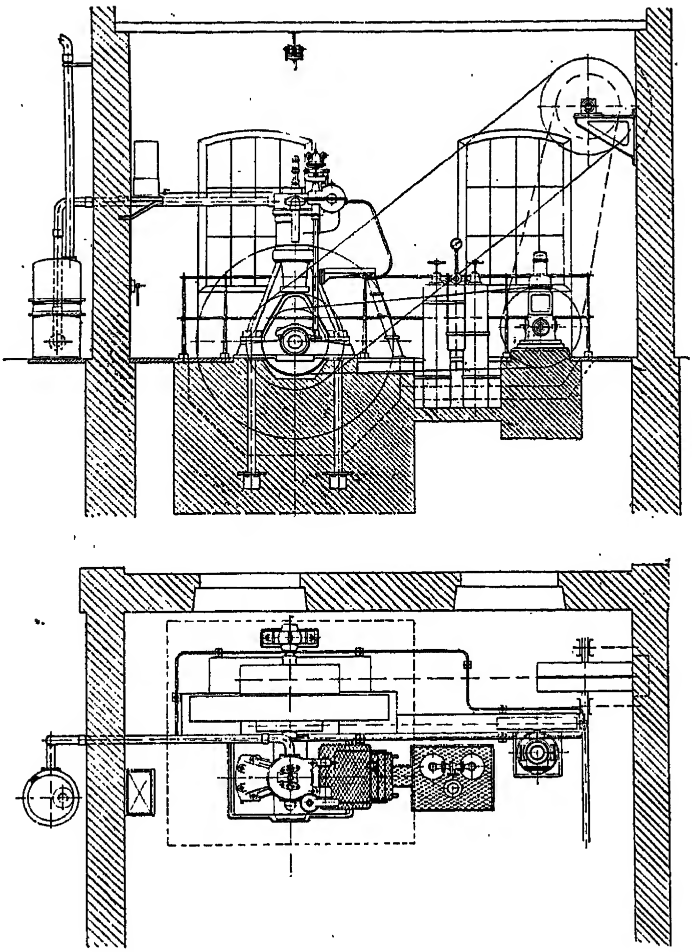
$$L = 2{\cdot}0 + 0{\cdot}025\,N + (n - 1)\,S,$$

in which N is again the B.H.P. per unit, n is the number of cylinders in the engine, and S is the distance between the centres of the cylinders. As found when dealing with crank shafts, S is equal to about 2·5 times the cylinder

* It is to be noted, however, that the diameter of the flywheel is usually greater than the dimension obtained from the formula, so that this must also be taken into account. In specifications, if the diameter of the flywheel is not given, the revolutions are given, facilitating thus the calculation of the diameter of the flywheel, it being known that the peripheral speeds generally in use are between 25 and 30 metres per second (80 to 100 ft. per sec.).

diameter, so that with close approximation the following values in metres may be given for S :—

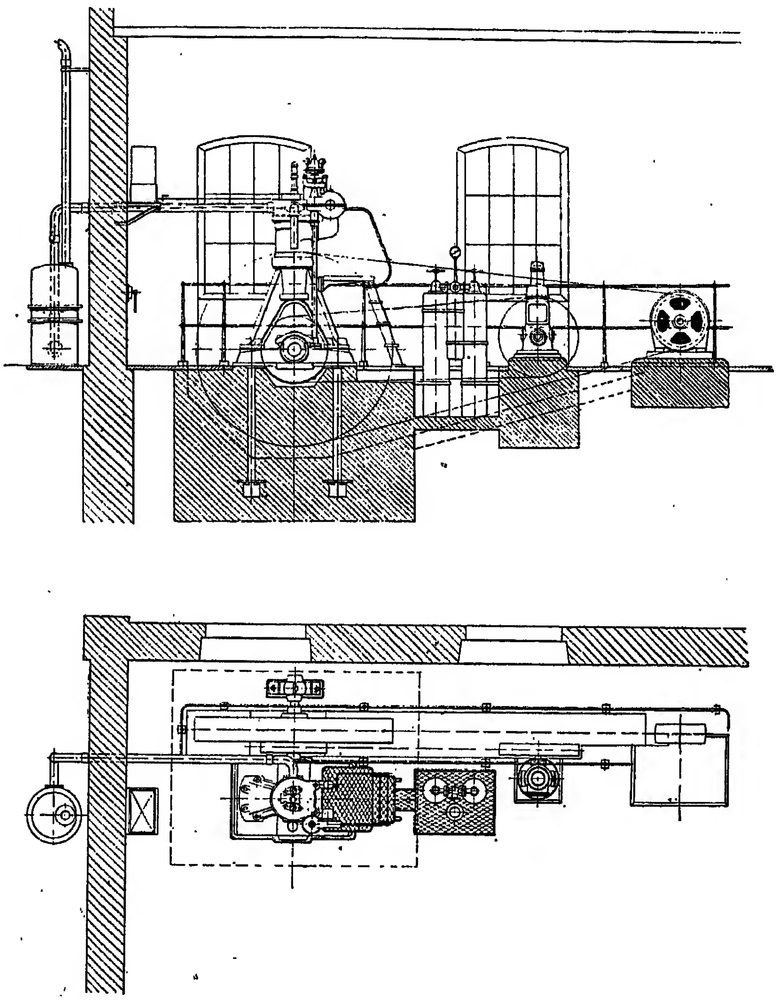
0·60 to 0·70	for cylinders of	30 to 36 B.H.P.
About 0·80	,,	40 B.H.P.
,, 0·90	,,	50 ,,
,, 1·0	,,	80 ,,
,, 1·10	,,	100 ,,



Figs. 329 and 330.—Single-cylinder Diesel Engine Plant driving a Transmission Shaft (Grazer).

This formula is only of use when the shaft carries the usual design of flywheel or pulley; for special shafts without pulleys or with several pulleys or with direct-coupled dynamos, the necessary deductions or additions must be made.

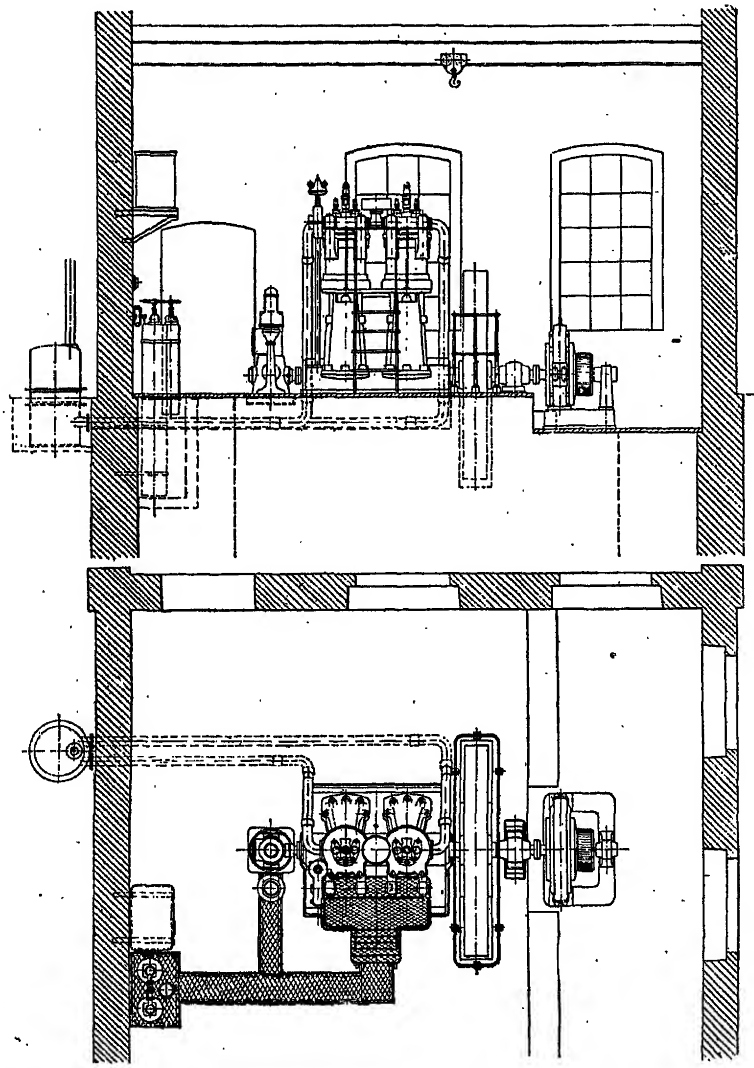
For special types of heavy oil engines it is impossible to give useful indications for determining the space occupied, on account of the large variety of designs on the market. Constructors, however, will always readily furnish outline drawings giving overall dimensions.



Figs. 331 and 332.—Single-cylinder Diesel Engine Plant driving a Dynamo (Grazer).

Foundations.—The plan of the foundation follows roughly the outline of the engine bed plate, being larger by some inches and extending to all the parts belonging to the engine, even when separate from the bed plate

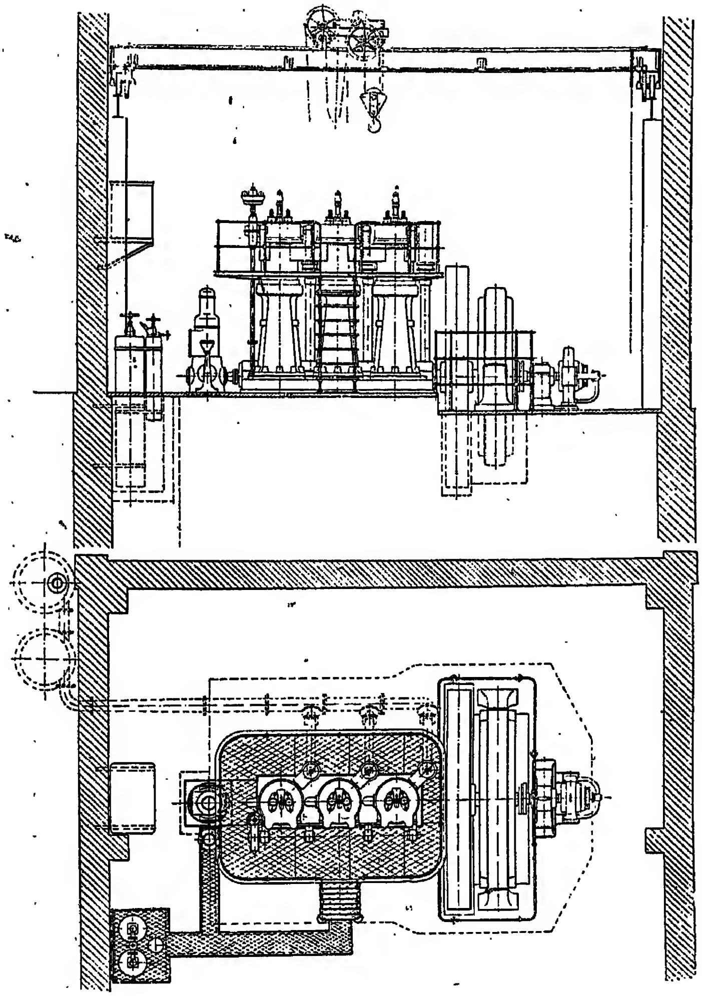
(*e.g.*, the outer flywheel bearing). The depth varies in different cases according to the nature of the soil, and where possible is carried down to solid ground.



Figs. 333 and 334.—Two-cylinder Four-cycle Diesel Engine Plant coupled direct to a Dynamo (Grazer).

Only with the smallest engines having no outer flywheel bearing is it sufficient to have a monolithic block of stone or concrete; usually the

foundation is of strong bricks and cement or mortar. The latter is the most suitable for damp ground, but usually a mixture of two or three parts of well-cleaned sand with one of Portland cement is adopted. For all those parts of the foundation above ground, and throughout if the foundation



Figs. 335 and 336.—Three-cylinder Four-cycle Diesel Engine Plant coupled direct to an Alternator (Grazer).

be very heavily loaded, the mixture should be richer—*i.e.*, equal parts of sand and cement. Stone is rarely substituted for bricks.

A cubic metre (35 cub. ft.) of foundation usually requires 400 bricks and from 260 to 300 litres (9 to 10·5 cub. ft.) of sand and cement.

In firm ground the foundation may be rectangular in form, but when the ground is yielding it is preferably formed with steps supported by a good layer of rubble from ·40 to 1·0 metre (1·3 to 3·3 ft.) deep, reinforced in particularly difficult places by good piling.

For four-cycle vertical Diesel engines in good ground the volume of the foundation reaches about 0·6 to 0·7 cubic metre (21 to 24·5 cub. ft.) per B.H.P., for single-cylinder engines, 0·45 to 0·50 cubic metre (15·8 to 17·5 cub. ft.) if the engine has two cylinders, 0·40 to 0·45 cubic metre (14 to

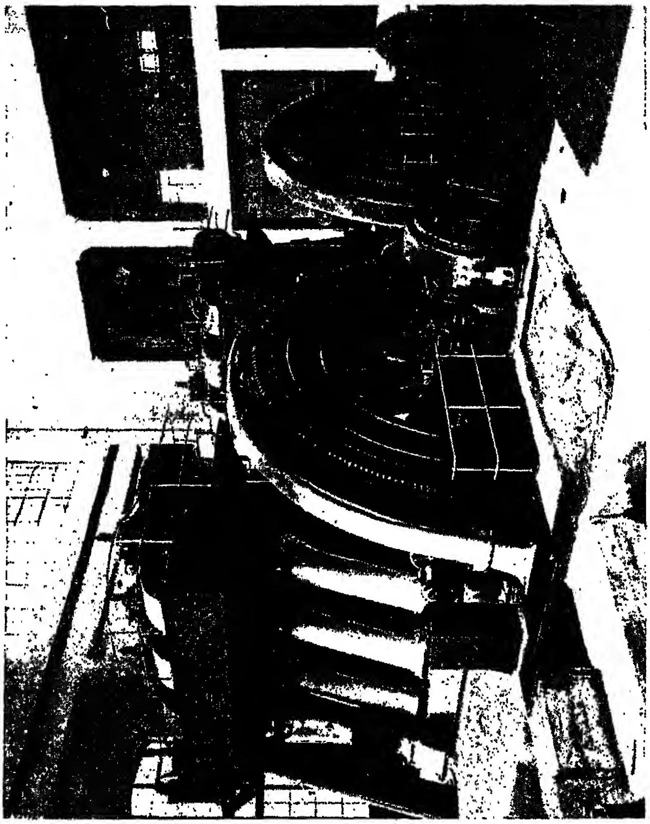


Fig. 337.—Central Electric Power Station at Rome. Three-cylinder Two-cycle Diesel Engines of 1,000 B.H.P. (Tosi).

15·8 cub. ft.) with three cylinders, and with four cylinders a little less than 0·4 cubic metre (14 cub. ft.), in each case per B.H.P. Horizontal engines have shallower foundations, but are larger in plan due to the larger engine bed plate; they are similar in every way to those for gas engines.

Usually the foundations are tunnelled to give access for tightening up the holding-down bolts, which, with Diesel engines, generally number $2(n + 1)$, where n is the number of cylinders, with 2 or 4 more to secure the outer flywheel bearing.

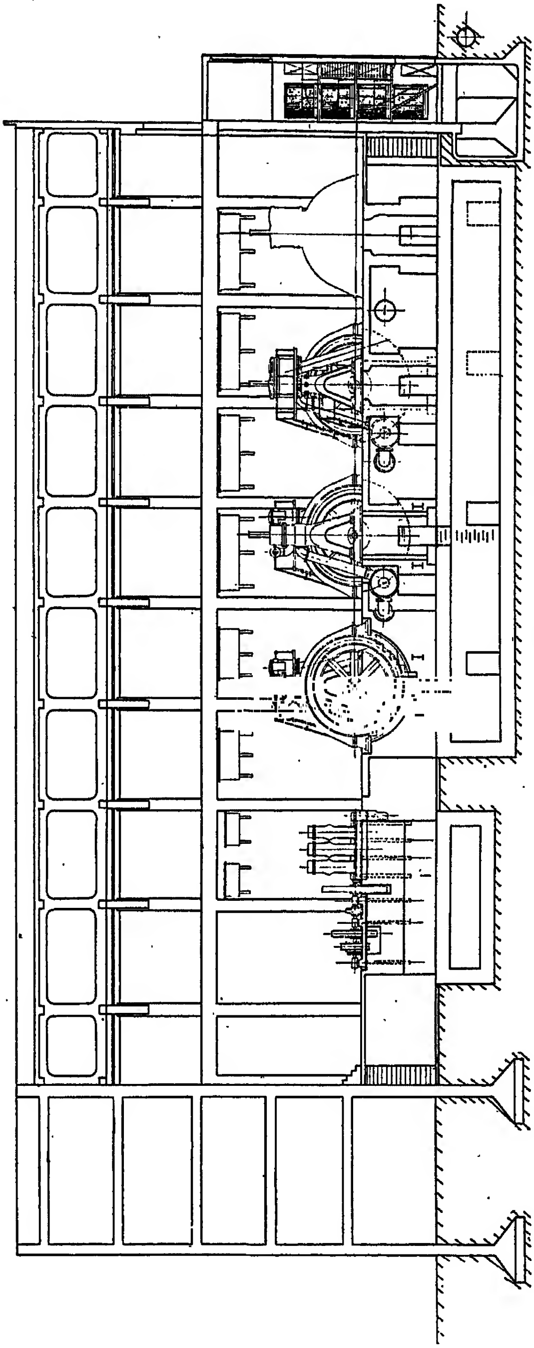


Fig. 338.—General View of Central Electric Power Station, Rome.

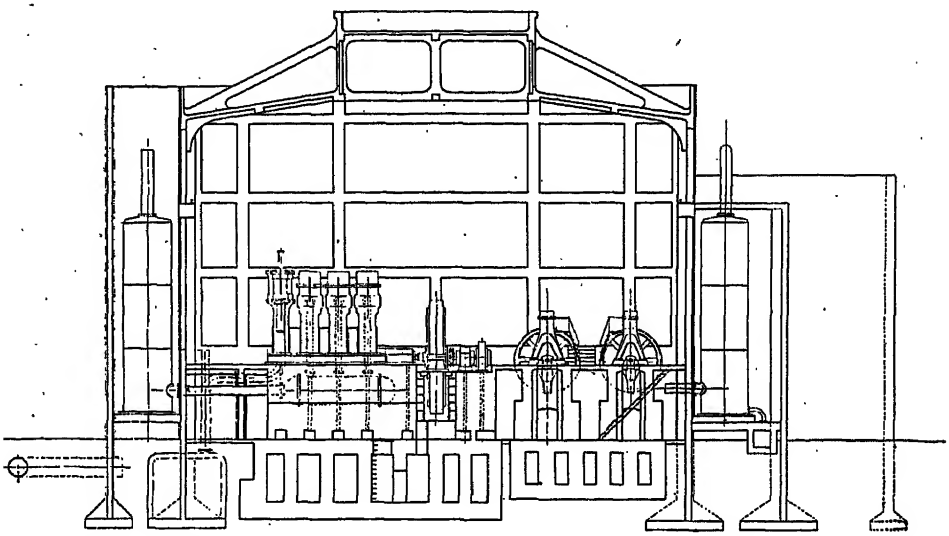


Fig. 339.—Central Electric Power Station, Rome. View showing Two-cycle Diesel Engine of 1,000 B.H.P. for driving Alternator, and Four-cycle Diesel Engines for driving Exciting Dynamos (Tosi).

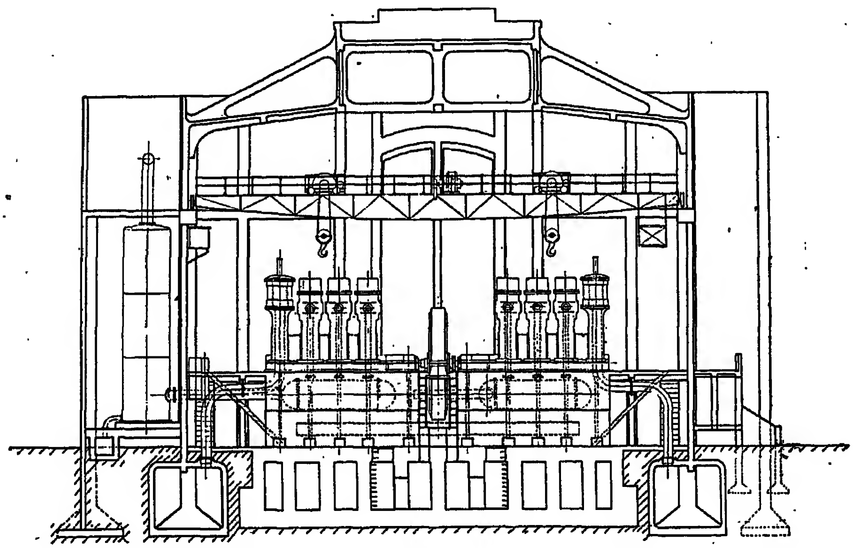


Fig. 340.—Central Electric Power Station, Rome. View showing Two-cycle Diesel Engine of 2,000 B.H.P. for driving Alternator (Tosi).

The bolts are generally secured at the bottom through a metal plate by means of a key or preferably a nut (Fig. 341); their length under the floor level is five or six times the cylinder diameter for vertical engines, and three to four times for horizontal engines.

For erecting and fixing the engine on its foundation the usual procedure is adopted; the bed-plate is placed on the foundation after the holding-down bolts have been put in position, the bed-plate and the outer flywheel bearing are carefully levelled up and thin chocks are fitted beneath them, after which liquid cement is run round the bed-plate flange and into the holding-down bolt holes, which are previously packed with broken bricks; finally the bolts are uniformly tightened up to prevent uneven and deforming stresses.

To make sure that this condition is satisfied, the crank is turned to near the top dead centre before the flywheel and piston are in place, and is allowed to fall. It should do so easily by itself and swing some distance past the bottom dead centre. In any case, after the bolts have been tightened up, it is well to bed the crank shaft again and scrape up the bearings.

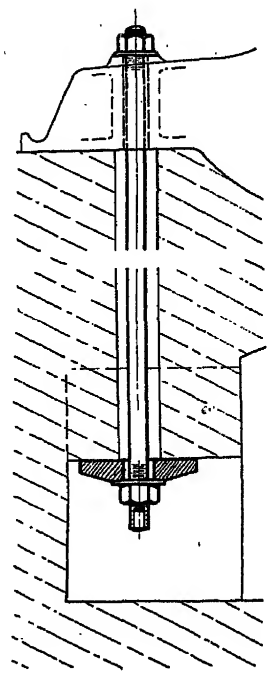


Fig. 341.—Method of securing Foundation Bolts.

Accessories.—Mention has already been made in the preceding chapters of many of the accessories, and important amongst those so far omitted for vertical engines is the starting platform. Since almost all the manœuvring levers are placed at the top of the engine close to the cylinder heads, an upper platform with a ladder is necessary, if the engine is over a certain height, in order to provide easy access to the fuel injection pumps and the starting levers.

There is nothing special about the construction of the platform; it may be made of angle irons and floor plates or of a thin plate of cast iron in one or more pieces. Sometimes the platform extends along the front of the engine only; this is sufficient for the operations of starting, etc.: for removing the cylinder heads, the valves, and for taking indicator diagrams it is better that it should be extended completely around the engine, as is the practice of some constructors.

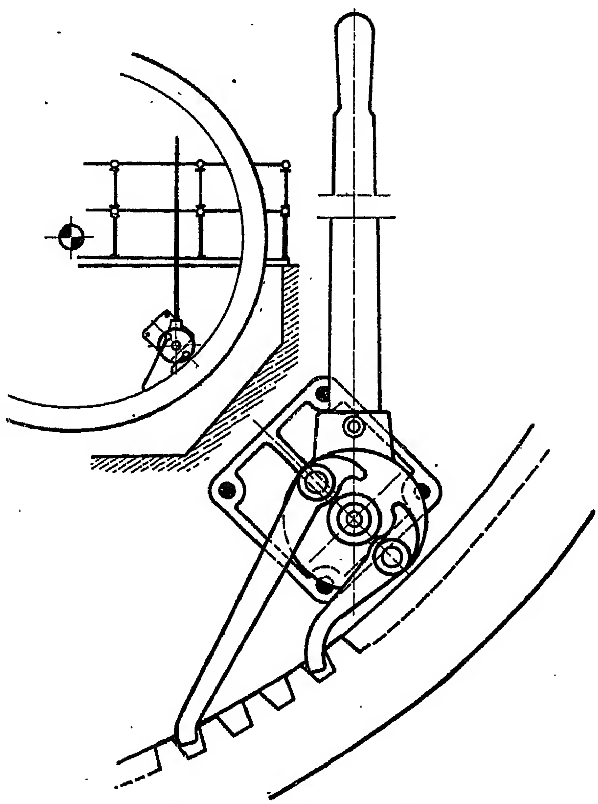
The width varies generally between 0·6 and 1·0 metre (2 to 3·3 feet), and its form is clearly seen in Plates IX. and XI., facing pp. 192 and 224; it is bolted on to the engine frames (*d*, Fig. 85, p. 81), and if it is not too large, projects with no other support than that of the ladder.

In very large plants consisting of several engines communicating bridges are sometimes provided, making it possible to pass from the platform of one engine to that of another.

Another accessory which is always necessary in vertical engines is the lifting gear for removing the piston. This consists of a set of chain blocks attached to a traveller plumbing the centre of the cylinder or cylinders

(Figs. 329 and 333, pp. 248 and 250). This apparatus is indispensable for the removal and replacement of cylinder heads and pistons, and its safe load should be high, in view of the weight of these parts. With very large plants a travelling crane is desirable (Figs. 335 and 340, pp. 251 and 254).

Figs. 342 and 343 show one of the commonest forms of hand-turning gear, which serves to turn the engine to the starting position when the strength of a man is not sufficient to move the flywheel direct. It engages with teeth formed on the inner circumference of the flywheel. When the



Figs. 342 and 343.—Typical Form of Hand-turning Gear.

flywheel does not serve as a belt pulley, these teeth may be formed on the outer periphery, in which case a lever may be used instead of the turning gear.

An essential requirement of the turning gear is that the pawls must be prevented from engaging when the engine is running.

Piping.—For a Diesel engine plant the piping may be classified as that for fuel, compressed air, cooling water, exhaust, and air suction.

The air for combustion may be drawn from the engine-room or from outside. The first method is by far the most widely adopted, and in addition

to the advantage of greater simplicity, it contributes efficiently to the ventilation of the engine-room; the second obviates the noise of the suction, which can never be entirely eliminated, although efforts are always made to reduce it to a minimum.

The general form of suction pipe drawing air from the engine-room is shown in Fig. 344. A vertical steel tube, closed at the bottom and of a length equal to five or six times its diameter, is attached to the under side of a cast-iron bend bolted to the cylinder head. This tube is perforated by a large number of narrow slots, through which the air is drawn, and practice has shown that this form of orifice is efficient in reducing noise.*

Instead of these tubes, some constructors substitute pipes without slots,

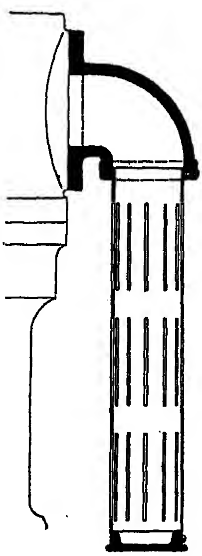


Fig. 344.—Suction Silencer.

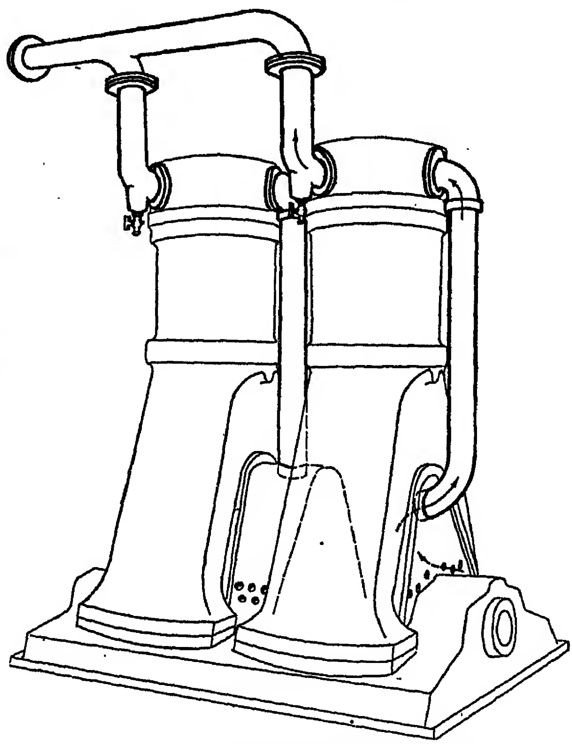


Fig. 345.—Arrangement of Suction from Crank Case.

but open at the end and drawing the air from the crank case (Fig. 345). By this means the advantage is obtained of drawing away, together with the suction air, the unpleasant smelling lubricating oil vapour, which, especially in the event of excessive lubrication, is given off by the pistons. When the crank case is of the form shown in Fig. 345, the sheet-steel casings are

* A current of air traversing narrow cuts makes much less noise than it would make in passing with the same velocity through round holes; when round holes are adopted it is difficult to avoid an unpleasant whistling sound.

perforated with a number of holes of sufficient diameter to permit the free entry of the air.

It is necessary to insure that the velocity of the suction air shall not be excessive, otherwise it will carry away a large quantity of the oil which escapes from the bearings under forced lubrication. With an engine on which the author carried out experiments, this phenomenon was so marked that, if the fuel pump were shut off after the engine had been running under load for a period (the cylinders consequently being hot), it continued to run light, burning only the lubricating oil drawn in with the air on the suction stroke.

The exhaust pipe for small engines is made of gas piping, and of ordinary cast-iron pipes for other engines. The piece next to the engine is always of cast iron, the better to withstand the high temperature of the exhaust gases. In engines of any considerable power water-jacketing is preferable.

The exhaust pipe may have the appearance shown in Fig. 345, with the first part vertical and leading into a horizontal pipe traversing the engine-room above the engine; or it may lead downward at the back of the frames to a little below the floor level and pass out of the engine-room through a tunnel (Plate IX., facing p. 192). When this second arrangement is adopted, the part above ground should be insulated or water-cooled, since without these precautions the hot pipes radiate an uncomfortable amount of heat.

With multi-cylinder engines, a collector pipe of diameter equal to, or rather larger than, the exhaust pipe of a single-cylinder engine of the same total power connects up the various exhaust branches.

It is good practice to fit drain cocks or small doors on the exhaust pipes close to each cylinder head, by means of which it is possible to know exactly in which of the cylinders, if in any, the combustion is defective.

The exhaust pipe discharges into one or more silencers, generally of cast iron, similar in every way to those for gas engines; the volume should be 10 or 20 times that of the sum of the cylinder volumes of the engine, with the inlet and discharge flanges at right angles to one another. A vertical pipe, which serves as a chimney, leads from the silencer and carries the exhaust gases somewhat higher than the roof of the factory. With large plants, chimneys of brick are sometimes adopted.

The open end of the exhaust discharge is designed so that the speed of the gases on escaping should be 20 to 25 metres (65 to 82 feet) per second; the suction pipes should have practically the same dimensions.

The fuel piping is frequently made of copper, where it is close to the engine, and where bends are frequent and a certain neatness is desirable; for the remainder gas piping is used.

During cold weather the fuel oil may become dense when its resistance to passage in the piping is very much increased. For this reason it is desirable to provide fuel suction pipes of large diameter, so that sufficient oil may reach the pump when the engine is being started. When the engine is warm its radiation renders the fuel sufficiently fluid, especially if the precaution be adopted of leading the fuel suction pipe along and close to the exhaust pipe (Tosi), or if a length of it is wrapped round the exhaust pipe (Langen & Wolf).

Constructors adopting closed fuel injection pumps (Figs. 281 and 286, pp. 202 and 205), into which it is not possible to introduce paraffin on starting the engine, as can be done with pumps of the type shown in Fig. 274 (p. 197), fit a three-way cock on the pump, as described on p. 208, by means of

which it is possible at will to put the pump in communication with either the heavy oil reservoir or with a smaller one containing oil of lower density.

The heavy oil reservoir is usually built of sheet steel, and has a capacity of about 10 litres (2·2 gallons) per B.H.P. of the engine. It is placed in a corner of the engine-room at a height of about 6 feet above the fuel injection pump of the engine, and is fitted with an arrangement to indicate clearly the fuel level therein.

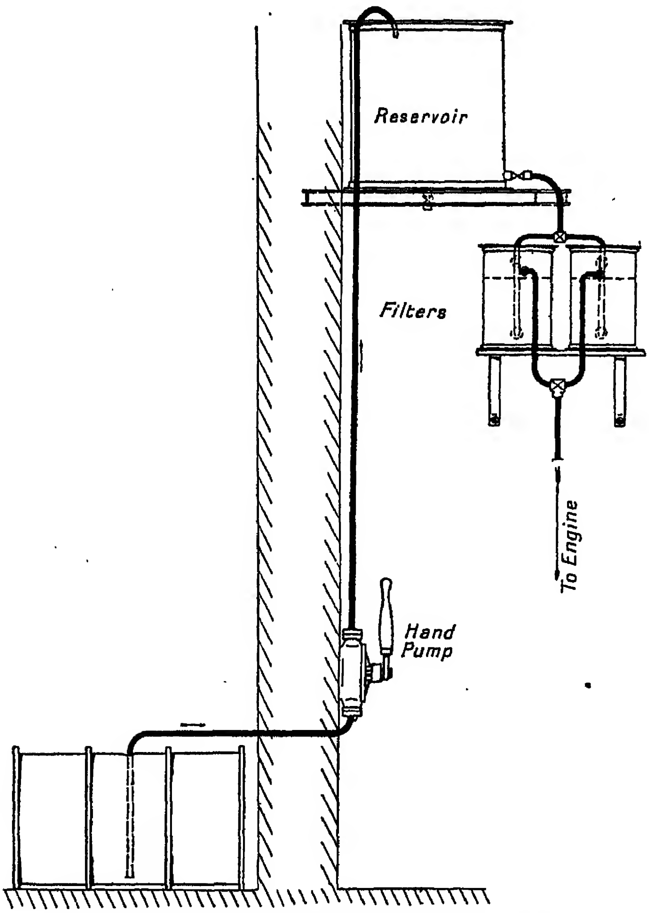


Fig. 346.—Arrangement of Fuel Supply to Engine.

With large installations an additional reservoir of sufficient dimensions for several months' supply of fuel is provided on the level of the floor or below the ground. This second reservoir is constructed of brick work or cement, lined with zinc or glass plates, in order to preserve the cement from the action of the oil. A hand pump delivers the fuel into the smaller reservoir from this main storage, or from the barrels in which it is received (Fig. 346).

On the pipe leading from the reservoir to the fuel injection pump on the engine are fitted two metallic gauze filters, each of which is sufficient to pass all the oil required for full load, so that one may be used when the other is being cleaned.

The cooling water of the engine should be clean, to prevent deposits in the spaces through which it circulates. It is desirable, if possible, not to use excessively hard water or sea water, which might form incrustations. If unavoidable, sea water may be used provided a sufficient quantity is circulated to ensure that the discharge temperature will not exceed some 40° C. (104° F.).

Incrustation hinders heat flow through the walls, and unless the engine be provided with special arrangements, its removal is difficult, and it is generally easier to dissolve it. In the case of calcareous deposits, a dilute acid solution is used (1 part of hydrochloric acid to 3 parts of water); this is allowed to act for 24 hours, after which it is necessary to pass a large quantity of pure water through the water-circulating spaces, to wash out the acid and to prevent corrosion.

The engine cooling water does not undergo any change in passing through the jackets, and may, therefore, be cooled and used again, with the addition thereto of some 10 per cent. per hour, to make up for evaporation and leaks.

Any of the well-known systems of steam engine condensers may be used to cool the circulating water; generally, cooling towers having steps, or similar apparatus (down which the water flows), or sprinklers, or simply shallow tanks of large area are adopted. There should always be a reserve of water sufficient for about an hour's running at full load.

Fig. 347 shows the general arrangement of the cooling water system of a Diesel engine plant. The pump *a* draws from the well or from the cooling tanks and delivers the water into the reservoir *b*, which should be of sufficient capacity to supply the engine for about half an hour's running, so that, whether the pump be driven directly or indirectly from the engine, water may be circulated and the jackets filled before starting, if, for purposes of examination or to prevent freezing, they have been emptied; thus examination of the pump and small repairs to it may also be made whilst the engine is running.

The reservoir *b* should be placed as high as possible,* and if its base is not at least 6 to 10 feet above the highest point of the engine water jackets the discharge should be arranged to have a low temperature, or steam may form and, owing to the insufficient head, may collect and stop circulation.

Two pipes lead from the reservoir, *c* from the bottom to the engine, and *d*, from the top serving as an overflow. The pipe *c* may branch into others of smaller diameter leading to the various jackets, giving thus a system of circulation *in parallel*.

More frequently, however, a system *in series* is adopted, in which all the water passes successively through the various cooling jackets. With multi-cylinder engines, of course, each cylinder has its independent circuit.

For series cooling, the general rule of leading the water inlet (*i.e.*, the coldest water) to the least heated parts of the engine and thence to those at higher temperature holds good. The following sequence is generally

* A minimum height of 18 feet above the floor level is often specified.

adopted :—Air cooler, air compressor, cylinder jacket, head, exhaust valve, and exhaust pipes where they are jacketed.

Frequently, however, the exhaust pipes are provided with a separate branch, in order to permit of independent discharges, one from the cylinder heads and the other from the exhaust pipe, making it possible to regulate the temperature of the water leaving the engine.

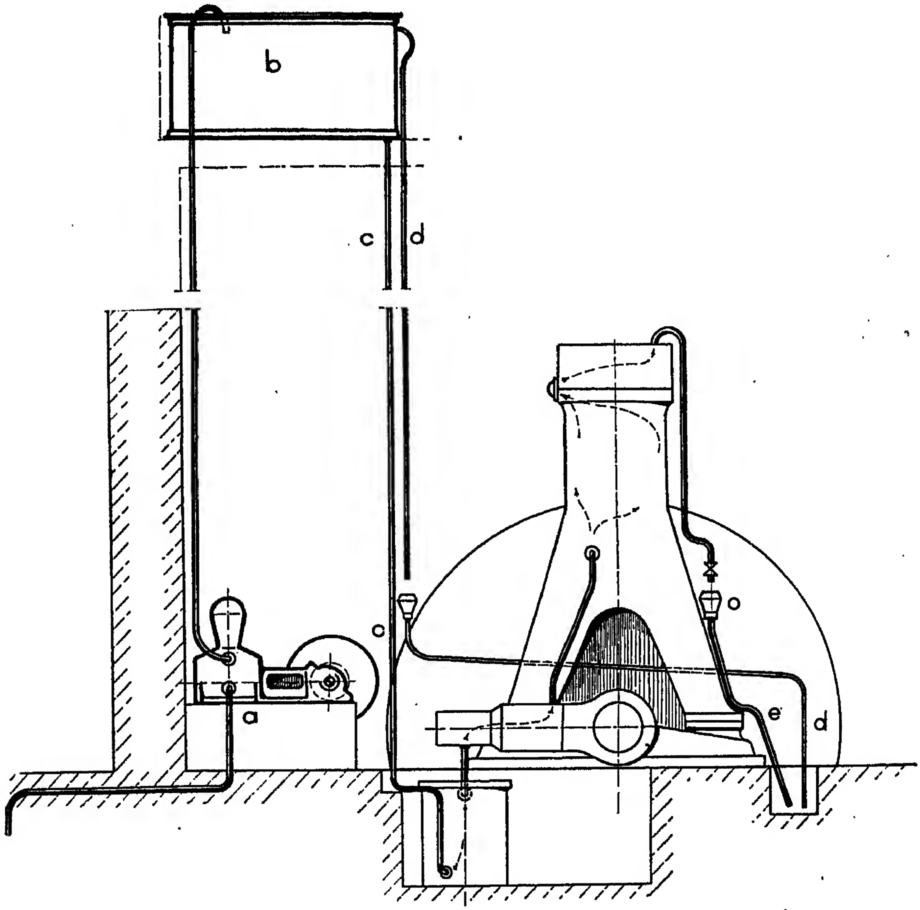


Fig. 347.—General Arrangement of Cooling Water System.

Regulation of the cooling water is desirable in order to attain the best conditions of running the engine.

When the circulation is under pressure from a pump the cooling water pipes of the cylinders, and their branches, if any, are each fitted with a cock at some point along their length. These cocks are often grouped at the ends of the pipes where they discharge into a common funnel *o* (Fig. 347).

This funnel leads through a pipe *e* of large diameter to the channel under the floor plates. In this way it is possible to regulate the delivery of the pipes at one point and to measure the various temperatures by the hand or with a thermometer.

When fresh-water pressure mains are available the plant remains unchanged, excepting that the pump and the reservoir *b* may be eliminated.

For the cooling water circuits gas piping is generally adopted, and the diameters should be such that the velocity of the water does not exceed 0·70 to 1·20 metres (2¼ to 4 feet) per second, for a quantity calculated on a basis of 20 litres (4·4 gallons) per B.H.P. The diameter of the pipe *e* should be one and a-half to twice that of the inlet *c*.

It is difficult to specify accurately the temperature of the discharge water. It varies for fresh water between 50° and 70° C. (120° and 160° F.), and depends on the part last traversed or on the order of the parts through which it has passed in series or in parallel, etc. It is certainly desirable that the compressor should be cold and the main cylinder hot, in order that these two parts may work efficiently. The cylinder head should be still hotter, to give readier ignition,* but the temperature of the cooling water spaces should never be so high that it is not possible to keep the hand thereon.

The hourly water circulation depends on the temperature of the discharge, and a statement that an engine consumes 12 or 15 litres (2·6 or 3·3 gallons) per B.H.P. per hour is of little value, if the difference of temperature between the inlet and discharge is not specified. It is known that the cooling water absorbs a given percentage of the total heat of the fuel—*i.e.*, 25 to 30 per cent.—so that this percentage should be found in the cooling water, or, in other words, the product of the quantity of water circulated, multiplied by the rise of temperature between the inlet and the discharge, should remain practically constant.

Differences in the quantities of water required to be circulated through various engines may generally be attributed to the smaller or greater number of parts cooled thereby; since, for example, when a considerable length of the exhaust pipe off the engine is water-cooled, a larger percentage of the total heat of the fuel will be carried away by the water, and so the quantity of the water required for circulation should be proportionally increased.

For practical purposes with engines of medium power, the required quantity is generally about 20 litres (4·4 gallons) per B.H.P.-hour, although, to give a suitable margin, the pump should be designed for a delivery of 30 to 40 litres (6·6 to 8·8 gallons) per B.H.P.-hour.

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* With gas engines the cylinder head should be kept cold to avoid the premature ignitions which, of course, do not occur with Diesel engines.

CHAPTER XV.

MARINE INSTALLATIONS.

The engine-room of a Diesel-engined ship is practically the same as that for the average steamship, in that the main engines are in the middle with the auxiliary machinery grouped at the back of the engine, or in small auxiliary machinery rooms in communication with the main engine-room.

The starting air reservoirs are generally arranged vertically against the bulkheads or horizontally in tiers, and the fuel injection air bottle is often placed conveniently close to the engine manœuvring gear.

As stated on p. 15, the engine-room of Diesel-engined ships is generally more roomy than that required for steam engines of the same power, since fewer auxiliaries are required and the necessary piping is considerably less, although the greatest care must be exercised to use the available space to the best advantage.

To give data for the space occupied would serve little useful purpose, as marine engines do not follow exactly definite types and no distinct classification is possible, owing to the differences between the engines of various makers for different spheres of action.

The auxiliaries of a Diesel-engined ship may be grouped under the headings of engines for working the ship, those required to manœuvre the ship, and those in connection with the main propelling units.

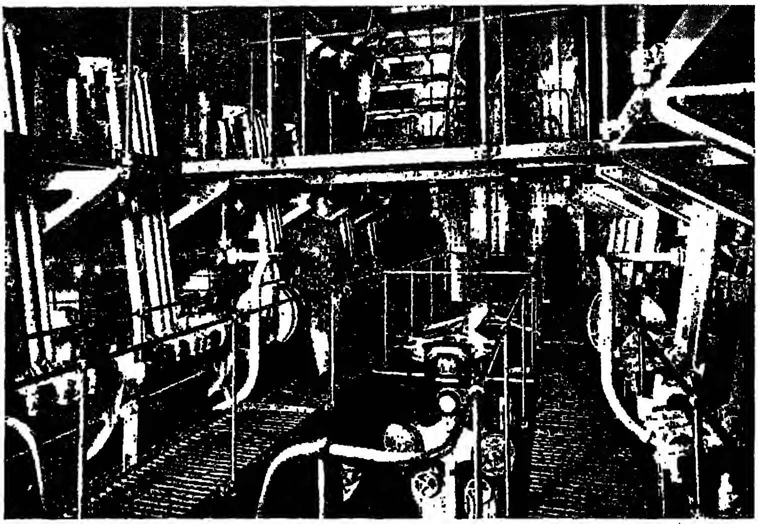
Of the first category, it is not necessary to treat at length, and briefly, these may be said to be the bilge, ballast, fire, sanitary, and fresh-water pumps, similar in every way to those in use on steam-engined ships. The same may be said of the refrigerating plant and the electric generators.

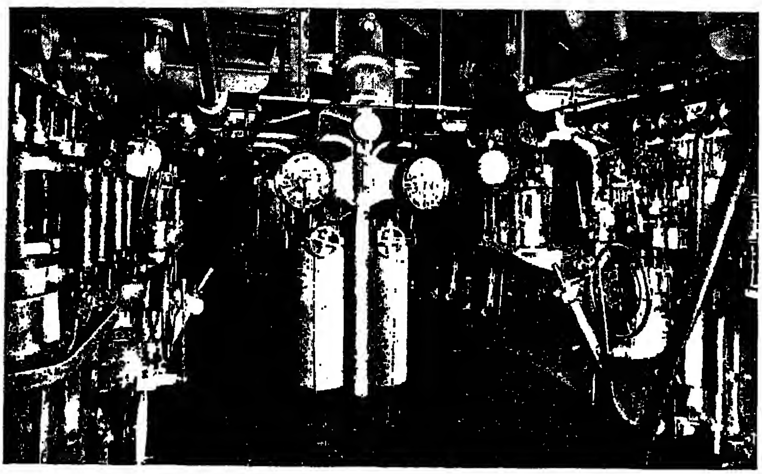
In cases where an auxiliary boiler is fitted, the dynamo may either be driven by the usual high-speed steam engine, or be coupled direct to a high-speed Diesel engine; in cases where the power is small, a petrol or paraffin engine suffices.

For large ships, two dynamo sets are generally provided, the main one being driven by a Diesel engine, and the other, which serves as a stand-by, being steam driven.

The cargo winches, as stated on p. 22, may be worked by steam or compressed air, and in either case they are exactly similar to the usual type. The same may be said of the warping and anchor capstans. The steering engine, if driven by compressed air, is generally supplied by a special reservoir charged by a separate compressor often driven from the main engine, and delivering air at a pressure of 7 to 8 atmospheres (100 to 115 lbs. per square inch).

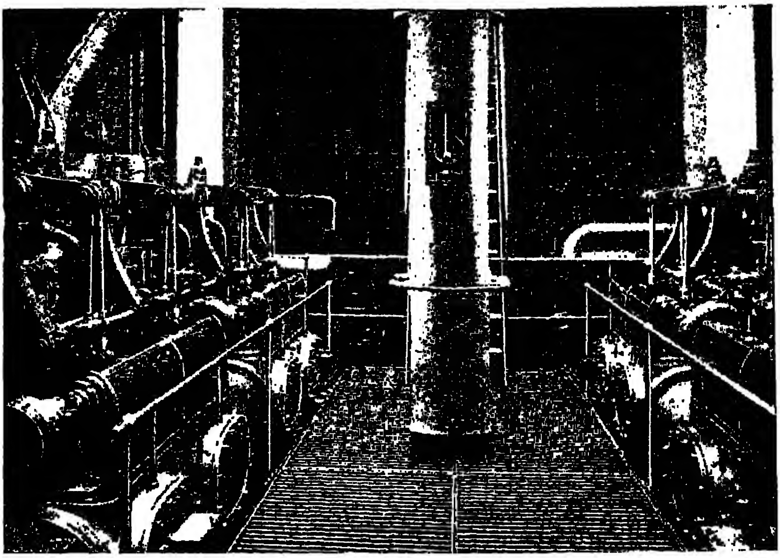
The auxiliaries for the main engines, in addition to the servo-motors necessary with high-power units for effecting the manœuvring of the main

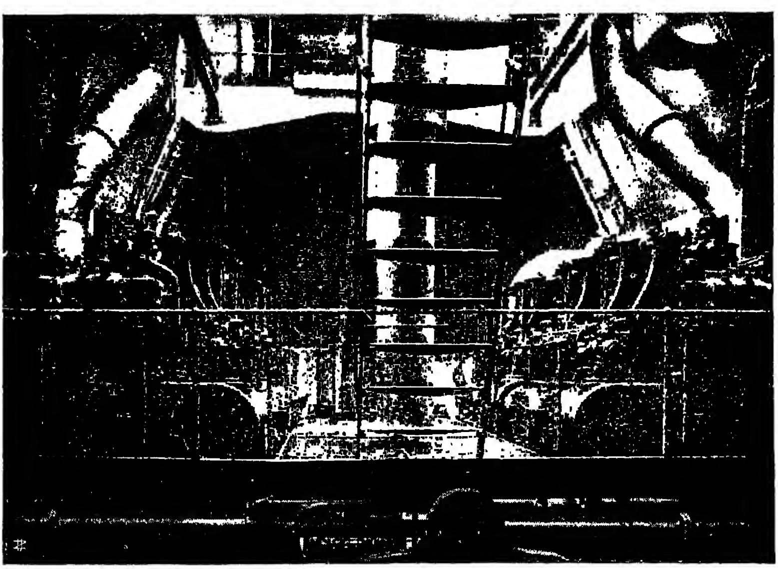




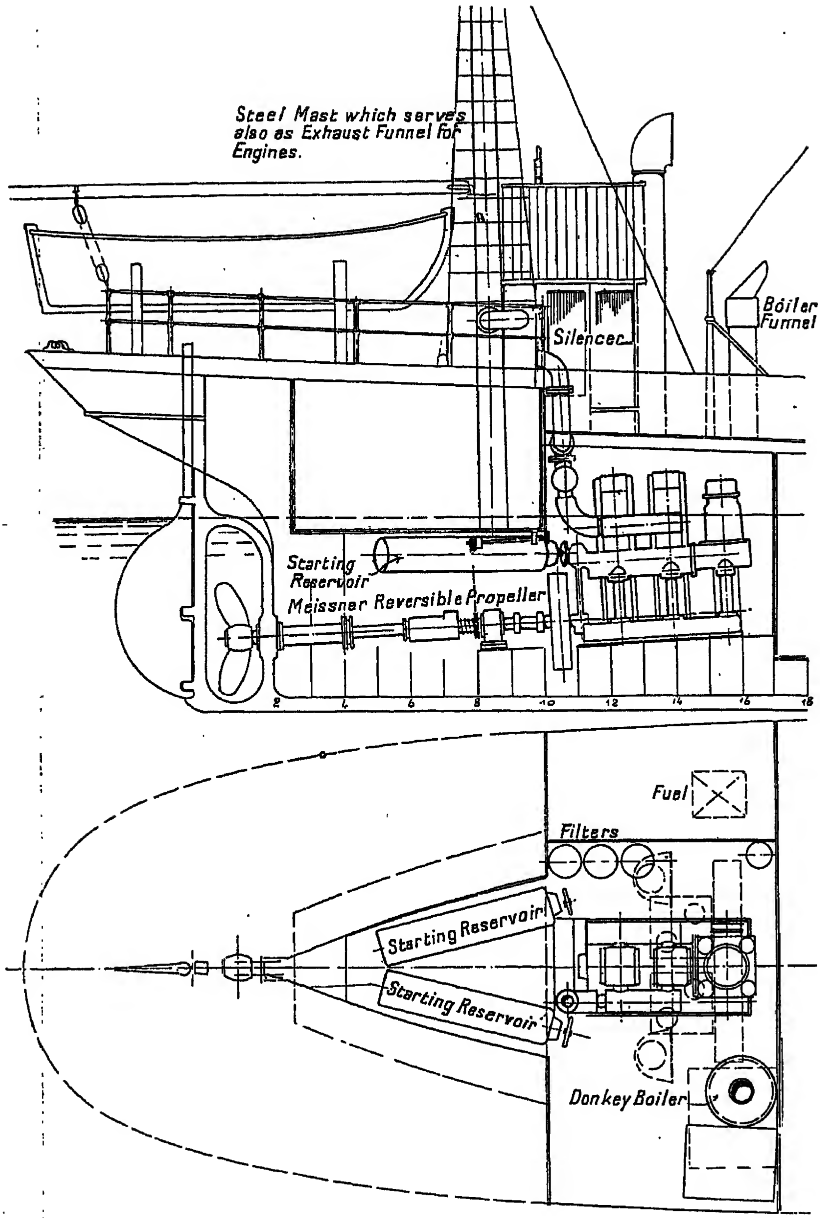
Figs. 348 and 349.—Engine-room of M.V. "Selandia."

engine, include the cooling water and fuel oil daily service pumps, the turning engine, and sometimes a reserve air compressor.



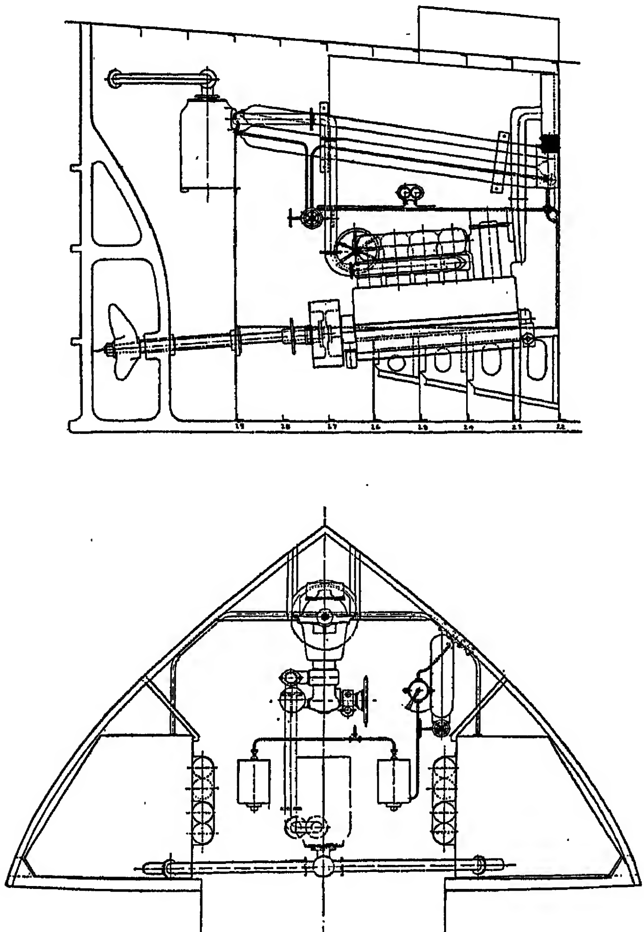


Figs. 350 and 351.—Engine-room of M.V. "Monte Penedo."



Figs. 352 and 353.—Arrangement of a Junkers-Frerichs Engine of 90 B.H.P. in the fishing-boat "Wotan."

The main engine cylinders are cooled by sea water supplied by a pump driven from the engine drawing from the sea and discharging generally into a reservoir placed at a height above the cylinder jackets in the engine-room coaming. From this height the water flows by gravity through the engine jackets and discharges overboard. For cooling the pistons fresh



Figs. 354 and 355.—A Two-cycle 150 B.H.P. Engine installation in the schooner "Aquila."

water is often applied, and on discharge from the piston is cooled and used again.

The delivery pressure of the cooling water pump is usually from 3 to 4 atmospheres (42 to 57 lbs. per square inch).

The reserve or stand-by air compressor is of the usual type, having two

or three stages of compression, and is similar to that coupled to the main engine, and of the same or slightly smaller capacity. In some cases two stand-by compressors are provided, of which one is driven by steam.

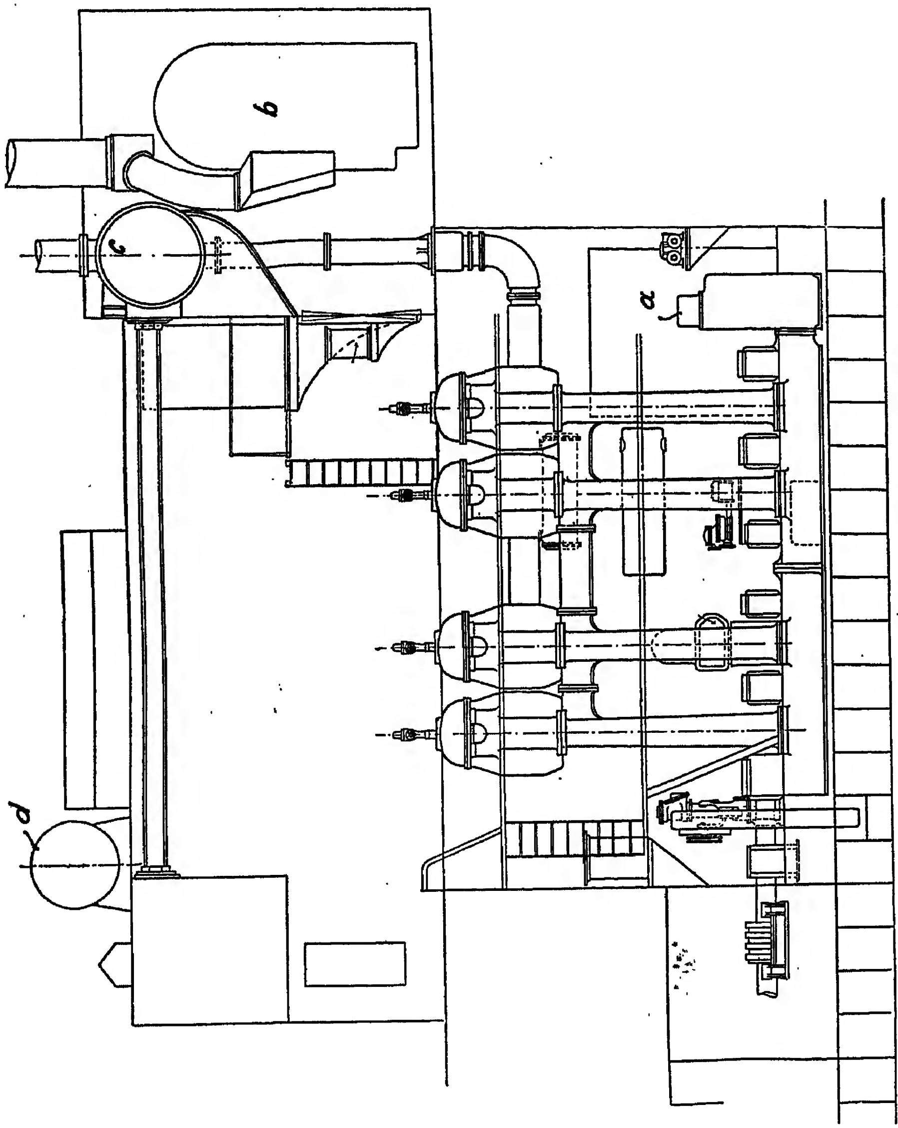


Fig. 356.—Engines of M.V. "Eavestone" (see Fig. 358).

Installations including a donkey boiler require the usual steam auxiliaries, such as the feed pump, surface condenser, etc.

The turning gear, which is generally arranged to engage with teeth in the periphery of the flywheel, may be actuated by hand or driven by steam, compressed air, or electricity.

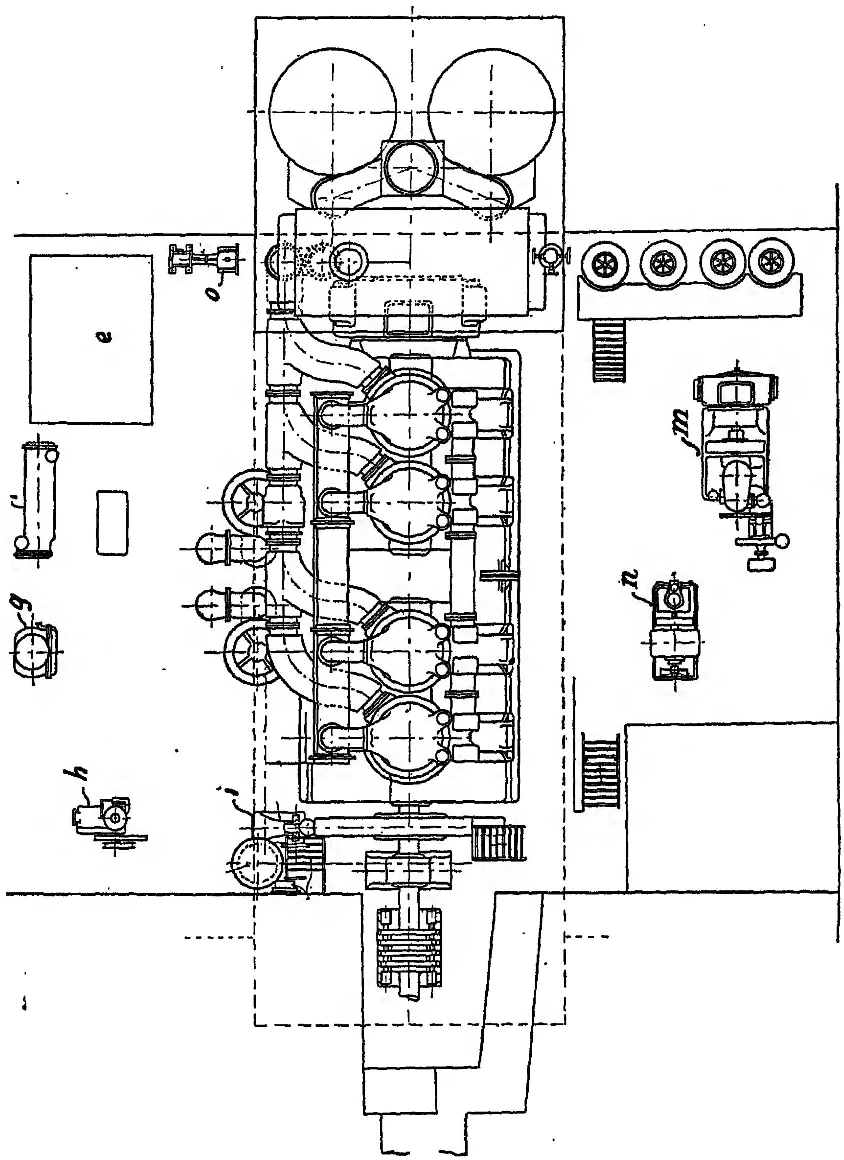


Fig. 357.—Engines of M.V. "Eavestone" (see Fig. 358).

The Diesel engines driving auxiliary machinery, such as compressors and dynamos, are generally of the high-speed type running at from 300 to

400 revolutions per minute; in almost all cases they work on the four-stroke cycle, and are identical in their main features with those constructed for the propulsion of small ships and for high-speed stationary work.

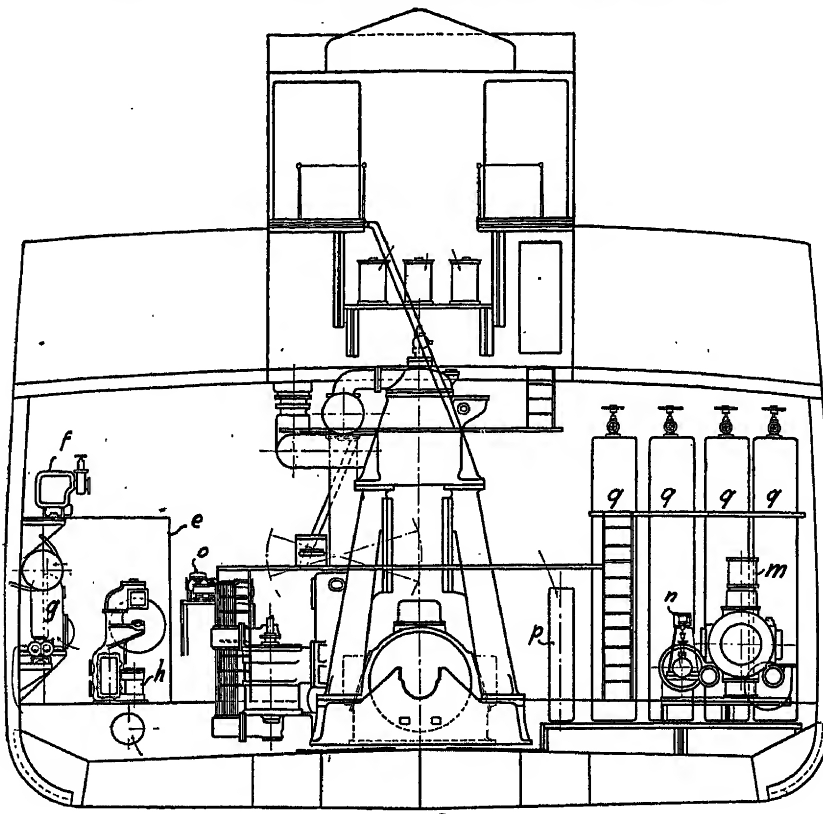


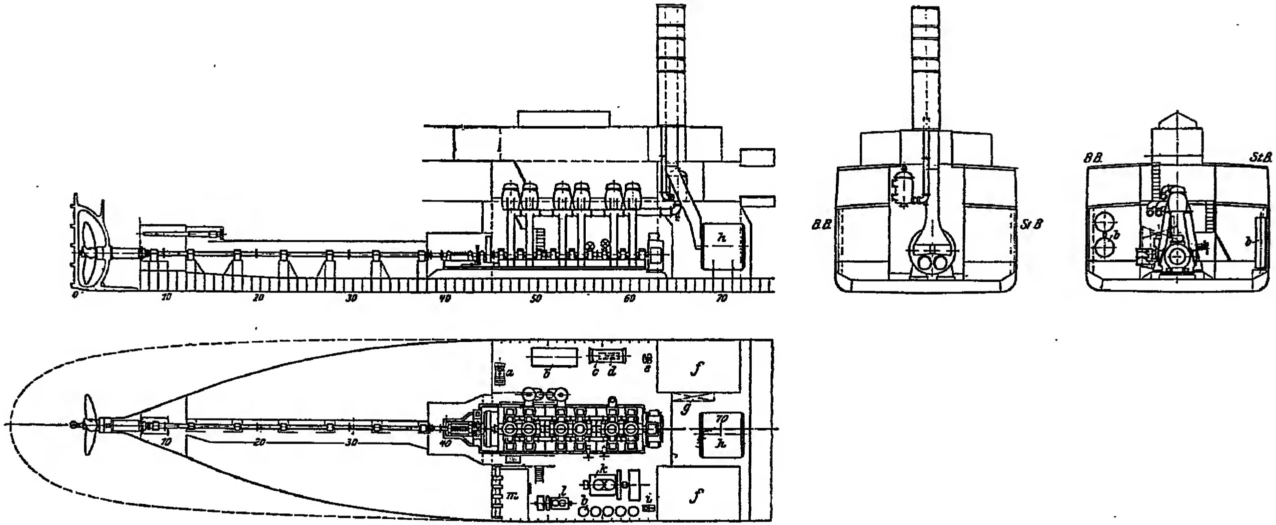
Fig. 358.

Figs. 356 to 358.—Four-cylinder Two-cycle Engine of 800 B.H.P. in the cargo M.V. "Eavestone."

e = Fuel reservoirs.
f = Auxiliary steam condenser.
g = Evaporators.
h = Steam-driven pump.
m = Auxiliary steam air compressor.
n = Steam-driven dynamo engine.
o = Steam boiler feed pump.
p = Fuel injection air reservoir.
q = Starting air reservoirs.

When dynamos have to be driven, flywheels of a sufficient size must be fitted to give the requisitely small degree of irregularity of running, about 10 per cent., and where two dynamos are installed arrangements are generally made to regulate the speed of revolution of the engines, in order that the two sets may be coupled in parallel.

In some cases, in order to reduce weight and first cost, it is arranged that one auxiliary Diesel engine may drive several auxiliary machines by the interposition of couplings or clutches.



Figs. 361 and 362.—Installation of a Two-cycle Diesel Engine of 1,500 B.H.P. in the M.V. "Rolandsek."

a = Electrically-driven bilge pump.
b = Starting air reservoir.
c = Condenser for steam-driven auxiliaries.
d = Pumps for same.
e = Steam-driven pump.
f = Fuel tank.
h = Auxiliary boiler.
i = Steam-driven air compressor.
k = Diesel-driven reserve compressor.
l = Diesel-driven dynamo.
m = Workshop.

The descriptions which have been given of the *suction, exhaust, and cooling-water piping* for stationary engines hold good generally for marine engines, with the exception of those cases in which the questions of space and weight

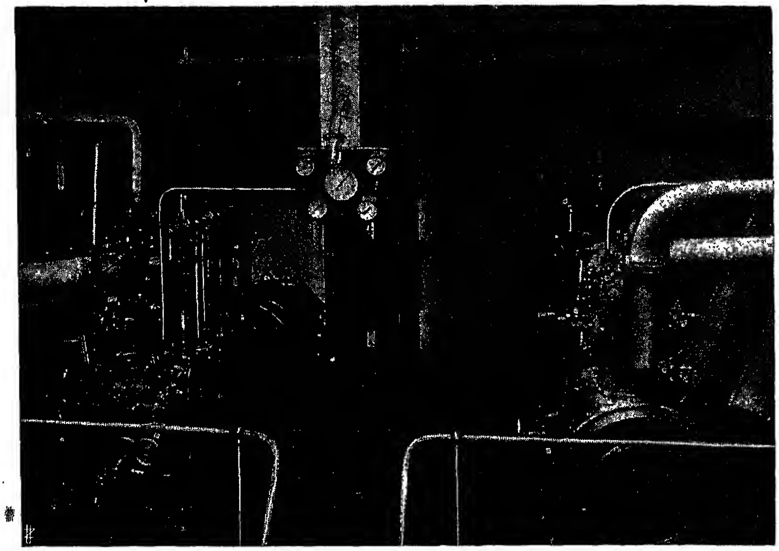


Fig. 363.—Reserve Compressor and Electric-light Dynamo driven by Diesel Engines on board the M.V. "Monte Penedo."

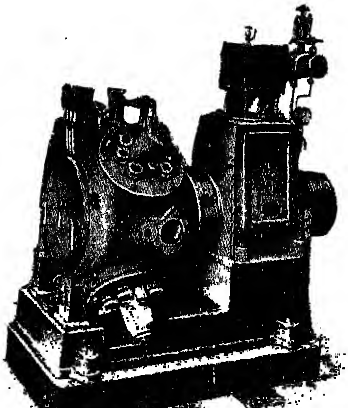
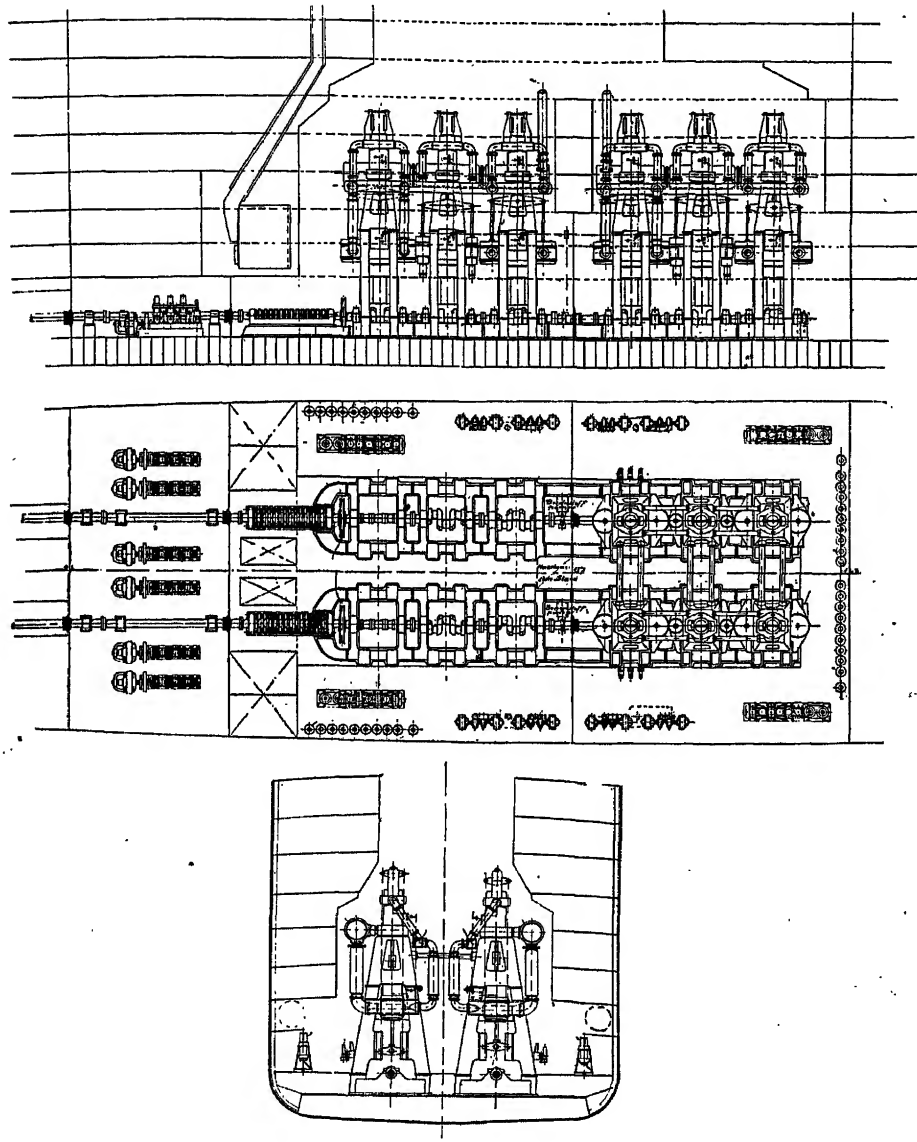


Fig. 364.—Steam-driven Reavell Compressor.

and the particular conditions to be satisfied require the adoption of other expedients. In the rules for the classification of marine Diesel engines, given in the Appendix to this chapter, suggestions are made in this connection.



Figs. 365 to 367.—Scheme of Installation of Two Junkers Engines of 30,000 total B.H.P. in an Atlantic liner af 36,000 tons displacement.

The engine seating consists of the framing to which the bed plate is bolted, and when being erected in place the bed plate should be levelled in the same way as already described for land engines.

APPENDIX TO CHAPTER XV.

RULES OF CLASSIFICATION SOCIETIES FOR THE CLASSIFICATION AND SURVEY OF MARINE DIESEL ENGINES.

The Marine Registration and Classification Societies have, for some time past, studied the application of the Diesel engine to marine propulsion, with a view to formulating rules for the construction and periodic survey of vessels equipped with this type of prime mover.

In view of the great number of different types of engines at present being built, and of the rapid nature of the development of this branch of engineering, the adaptation of rigid formulæ to bring within their scope the majority of the systems in vogue is extremely difficult, but the rules which have been drawn up, and which are reproduced in the following pages, have been evolved as a result of considerable experience with marine Diesel engines :—

LLOYD'S REGISTER OF SHIPPING.

NOTICE IS HEREBY GIVEN THAT THE FOLLOWING RULES FOR THE CONSTRUCTION AND SURVEY OF DIESEL ENGINES AND THEIR AUXILIARIES HAVE BEEN ADOPTED BY THE GENERAL COMMITTEE.

RULES FOR THE CONSTRUCTION AND SURVEY OF DIESEL ENGINES AND THEIR AUXILIARIES.

Section 1. In vessels propelled by Diesel Oil Engines, the Rules as regards machinery will be the same as those relating to steam engines, so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

CONSTRUCTION.

Section 2. 1. In vessels built under Special Survey and fitted with Diesel Engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel Engines being built under Special Survey, the distinguishing mark ✠ will be noted in Red, thus:—✠LMC or ✠NE.

3. In order to facilitate the inspection, the plans of the machinery are to be examined by the Surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full power working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lbs. per square inch, the diameters of the crank shaft of the main engines are not to be less than those given by the following formula :—

$$\text{Diameter of crank shaft} = \sqrt[3]{\mathbf{D}^2 \times (\mathbf{AS} + \mathbf{BL})},$$

where **D** = diameter of cylinder,
S = length of stroke,
L = span of bearings adjacent to a crank, measured from inner edge to inner edge.

The values of (**AS** + **BL**) are as given in the following table :—

TABLE I.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Values of the Coefficients.
4 or 6 cyls.	2 or 3 cyls.	·089 S + ·056 L
8 cyls.	4 cyls.	·099 S + ·054 L
10 or 12 cyls.	5 or 6 cyls.	·111 S + ·052 L
16 cyls.	8 cyls.	·131 S + ·050 L

For auxiliary engines of the Diesel Type the diameters may be five per cent. less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should be not less than 1·33 times and the thickness not less than 0·56 times the diameter of the shaft as found above, or, if these proportions are departed from, the webs must be of equivalent strength.

9. Where no flywheel is fitted, the diameter of the intermediate shaft must not be less than given by the formula :—

$$\text{Diameter of intermediate shaft} = \text{coefficient } \sqrt[3]{\mathbf{D}^2 \times \mathbf{S}},$$

where **D** = diameter of cylinder,
S = stroke of piston,

and the value of the coefficient is given by the following table :—

TABLE II.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Value of the Coefficient.
4 cyls.	2 cyls.	·456
6, 8, 10, or 12 cyls.	3, 4, 5, or 6 cyls.	·436
16 cyls.	8 cyls.	·466

Where the stroke is not less than 1·2 times nor more than 1·6 times the diameter of the cylinder, (·735 **D** + ·273 **S**) may be taken instead of $\sqrt[3]{\mathbf{D}^2 \times \mathbf{S}}$.

10. In cases where flywheels are fitted, the following value of the coefficient may be taken for determining the size of the intermediate shaft abaft the flywheel shaft.

TABLE III.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Value of the Coefficient.
4 cyls.	2 cyls.	·405
6 cyls.	3 cyls.	·400
8 cyls.	4 cyls.	·409
10 cyls.	5 cyls.	·420
12 cyls.	6 cyls.	·427
16 cyls.	8 cyls.	·461

11. The diameter of the flywheel shaft must be at least equal to that of the crank shaft.

12. The diameter of the thrust shaft measured under the collars must be at least $\frac{21}{20}$ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

13. The diameter of the screw shaft must be not less than the diameter of the intermediate shaft (found as above) multiplied by $\left(\cdot 63 + \frac{\cdot 03\,\mathbf{P}}{\mathbf{T}}\right)$, but in no case must it be less than 1·07 **T**,

where **P** = the diameter of the propeller in inches,
T = the diameter of intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 18, paragraph 3, of the Rules for Engines and Boilers. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be $\frac{21}{20}$ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

14. If the designed maximum pressure in the cylinders exceeds 500 lbs. per sq. inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{maximum pressure in lbs. per sq. inch}}{500}}$.

15. Where the cylinder liners are made of hard close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $\frac{1}{15}$th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

16. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lbs. per square inch, and must be perfectly tight at that pressure.

17. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

18. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent. above the designed maximum pressure in the cylinders and discharging where no damage can occur.

19. The air compressors and their coolers are to be made so as to be easy of access for overhaul and adjustment.

20. In single screw vessels, an auxiliary air compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the manœuvring gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin-screw engines in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high-pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

AIR RECEIVERS.

Section 3. 1. Compressed air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. *Quality of Material.*—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 4, par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent. less than that required with boiler plates.

5. *Tensile and Bend Tests* are to be made from the material of *each* receiver. When they are welded or riveted, the tests may be made, and the thicknesses verified, before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the

Surveyor before the ends are closed in, and at this time the Surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formulæ :—

Maximum working pressure in lbs. per sq. inch

$$= \frac{C \times S \times (T-2)}{D} \text{ for thicknesses of } \tfrac{5}{8} \text{ in. and above,}$$

$$= \frac{C \times S \times (T-1)}{D} \text{ for thicknesses below } \tfrac{5}{8} \text{ in.,}$$

where S = Minimum tensile strength of the steel material used, in tons per sq. inch,
T = Thickness of the material, in sixteenths of an inch,
D = Internal diameter of cylinder, in inches,
C = Coefficient as per following table :—

Coefficient			
77	for seamless receivers	of thickness	of $\frac{5}{8}$ in. and above
69	,, ,,	,, ,,	below $\frac{5}{8}$ in.
54	,, welded	,, ,,	of $\frac{5}{8}$ in. and above
48	,, ,,	,, ,,	below $\frac{5}{8}$ in.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T = \frac{D}{17} \times \sqrt{P},$$

where T = thickness, in sixteenths of an inch,
D = internal diameter, in inches,
P = working pressure, in lbs. per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates is to be tested by hydraulic pressure to twice the working pressure for pressures up to 200 lbs. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lbs. per square inch above the working pressure.

12. All receivers above 6 inches internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air compressor

is fitted with a safety valve so arranged and adjusted that no greater pressure than permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

PUMPING ARRANGEMENTS.

Section 4. 1. The requirements of the pumping arrangements for the various holds, double bottoms or other ballast tanks, etc., are to be the same as required in steam vessels of the same size.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled while the other is at work. In twin-screw vessels one bilge pump upon each engine will be approved. These pumps are to be arranged to draw from all compartments. Independent power-driven pumps may be fitted in lieu of these, if desired.

3. A steam pump, or equivalent power-driven pump, is also to be provided with connections to enable it to draw from all compartments and from the sea. It must be arranged to discharge overboard and also on deck to the fire service pipes. It must have at least one suction to the engine room bilge distinct from those connected with the bilge pumps, so that it may be used for pumping from the engine room when the bilge pumps are being used upon other parts of the vessel.

4. In addition to the above, where water ballast is used, the water ballast pump must have one direct suction from the engine-room bilges. (This is in lieu of the bilge injection required with steam engines.)

GENERAL.

Section 5. 1. For the ordinary fuel tanks the requirements of Section 49 will apply. The daily service and other separate tanks must be tested, with all their fittings, with a head of water of 12 feet above their highest points. They must be fitted with air pipes discharging above the upper deck. If they are fitted with glass gauges for indicating the quantity of oil contained in them, arrangements must be made for readily shutting off the gauges in the event of the breakage of the glass, and for preventing any damage from leakage of oil.

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for cables for electric light.

4. It is recommended that all pipes conveying fuel oil should, as far as possible, be made of steel or iron, rather than copper, owing to the rapid corrosion of copper pipes when using oil containing sulphur.

SPARE GEAR.

Section 6. The articles mentioned in the following list will be required to be carried, viz. :—

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, etc., fitted to it.

In addition, one complete set of valves, valve seats, springs, etc., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of cylinders of each engine.
1 piston complete, with all piston rings, studs, and nuts for the main engines.
In addition, one set of piston rings for one piston of the main and of the auxiliary Diesel engines.
1 complete set of main skew wheels for one main engine.
2 connecting-rod, or piston-rod, top-end bolts and nuts, both for the main and for the auxiliary Diesel engines.
2 connecting-rod bottom end bolts and nuts, both for the main and for the auxiliary Diesel engines.
2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.
1 set of coupling bolts for the crank shaft.
1 set of coupling bolts for the intermediate shaft.
1 complete set of piston rings for each piston of the main and of the auxiliary compressors.
1 half set of valves for the main and for the auxiliary compressors.
1 fuel pump complete for the main engine, or a complete set of all the working parts.
1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.
1 set of valves for the daily fuel supply pump.
1 set of valves for the water circulating pumps.
1 set of valves for one bilge pump.
1 set of valves for the scavenge pump, where lift valves are used.
A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.
Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

PERIODICAL SURVEYS.

Section 7. 1. The engines are to be submitted to survey annually, and in addition are to be submitted to a Special Survey upon the occasion of the vessels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 19 of the Rules for Engines and Boilers.

2. *Special Surveys.*—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz. :—All the cylinders, pistons, valves and valve gears, connecting-rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes, sea connections and their fastenings, are to be examined. The air compressors are also to be examined. The air receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this Section.

3. *Annual Surveys.*—The whole of the parts of the engines which the

engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The Survey must include, for each main engine, the examination of at least 2 pistons, 2 cylinder covers and their valves, 2 connecting-rods and their brasses, both top and bottom ends, 2 of the main bearings and crank-shaft journals, and 1 of the tunnel bearings. If these are all satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air receivers cannot be examined internally, they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

The pumps and air compressors must be examined and tried under working conditions. If found to be satisfactory, they need not be dismantled.

The manœuvring of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. *Record of Survey.*—If the various parts mentioned in paragraph 2 or 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. *Survey of Screw Shafts.*—Screw shafts are to be drawn and examined at intervals of not more than two years.*

NOMINAL HORSE-POWER OF DIESEL ENGINES.

The following rule is to be used for determining the Nominal Horse-Power of Diesel Engines in regulating the fees for their survey, viz. :—

$$\text{NHP} = \frac{N \times D^2\sqrt{S}}{80}$$ in the case of single-acting engines of the 4-cycle type,

$$= \frac{N \times D^2\sqrt{S}}{40}$$ in the case of single-acting engines of the 2-cycle type, and

$$= \frac{N \times D^2\sqrt{S}}{20}$$ in the case of double-acting engines of the 2-cycle type,

where D = diameter of cylinder in inches,

S = stroke of piston in inches in ordinary reciprocating engines.

= twice the stroke of piston in the case of engines of the "Junker" type,

N = number of cylinders.

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* On the application of Owners, the Committee will be prepared to give consideration to the circumstances of any special case.

For the survey and testing of steam boilers fitted in Diesel-engined vessels additional fees will be charged in accordance with the Society's usual scale.

It is to be understood that the following Rules of

THE BRITISH CORPORATION FOR THE SURVEY AND REGISTRY OF SHIPPING

are, in the fullest meaning of the word, Provisional, and form in effect a basis for discussion with Designers when machinery of this type is about to be built to the Classification of this Society.

PROVISIONAL RULES FOR INTERNAL COMBUSTION ENGINES.

General Conditions of Classification.

The construction of Internal Combustion Engines, their auxiliaries and donkey boilers, which are intended for classification with the British Corporation, is to be carried out under the supervision and to the satisfaction of the Surveyors, and before proceeding with the work, full information regarding the engines, together with detailed plans and particulars of proposed arrangement of engine seating and holding-down bolts, bilge and ballast pumping arrangements, air reservoirs, oil fuel tanks, filters and oil warming arrangements, also detailed plans and particulars of boilers, showing their position on board ship, are to be submitted for the approval of the Committee. Upon satisfactory completion in accordance with the Rules, the machinery will be entered in the Register Book M.B.S. * (" Machinery British Standard "—Special Survey.)

The following Rules are intended to apply to the particular requirements in connection with the construction of Internal Combustion Engines, with their auxiliaries, air reservoirs, and oil fuel tanks, and are supplementary to such of the general requirements in connection with forgings, pumps, etc., for which provision is made under Section 33, etc., of the Rules.

(1) All material used in the construction of the machinery is to be made in accordance with Sec. 33, pars. 2-14. All steel for air reservoirs is to be subject to the requirements for boiler steel (Sec. 32), and that for oil tanks to those for ship steel (Sec. 3).

(2) The engine room is to be thoroughly ventilated and so arranged as to prevent the accumulation of inflammable gases. Satisfactory provision is to be made for the interception of any overflow of fuel from the engines. The fuel tank for the daily supply is to be kept well away from the donkey boiler, the funnel, and the exhaust pipes. Where fuel of a lower flash point than 150 deg. Fahr. is used, donkey boilers must be placed in an efficiently ventilated compartment separate from the engine room and fuel tanks.

(3) Propelling engines are to be fitted with a governor, or other satisfactory arrangement, to prevent racing of the engines. Engines above 300 B.H.P. should be reversible.

(4) **Shafting.**—The minimum diameter of crank, intermediate, thrust, and propeller shafts may be found from the following formulæ :—

$$\text{For crank shafts, } d = \sqrt[3]{\frac{D^2 \times P\,(S + L)}{C}}$$

where d = Diameter of shaft, in inches,
D = Diameter of cylinder, in inches,
S = Stroke of piston, in inches,
L = Length between edges of bearings, in inches,
P = Maximum initial pressure, in lbs., per square inch,
C = Constant.

C	For 4-cycle Engines, Single Acting. No. of Cyls.	For 2-cycle Engines, Single Acting. No. of Cyls.	For 2-cycle Engines, Double Acting. No. of Cyls.
9,000	1, 2, 3, 4	1, 2, 3	1, 2
8,500	6	4	3
8,000	8	...	...
7,500	12	6	4
7,000	...	8	...
6,500	...	...	6
6,000	...	6*	6*
4,500	...	8*	...

$$\text{For intermediate shafts, } d = \sqrt[3]{\frac{D^2 \times P \times S}{C}}$$

C	For 4-cycle Engines, Single Acting. No. of Cyls.	For 2-cycle Engines, Single Acting. No. of Cyls.	For 2-cycle Engines, Double Acting. No. of Cyls.
11,000	1, 2, 4, 6	2	...
8,750	12	1, 3, 6	1
7,800	3, 8	4	2, 3
6,700	...	8	...
6,200	...	...	4
4,800	...	6*	...
4,350	...	...	6
4,250	...	...	6*
3,750	...	8*	...

Thrust shafts to be 5 per cent. larger in diameter than the intermediate shafts.

Propeller shafts to be of the diameter of the Thrust Shaft, multiplied by the value of B in the Table for Propeller Shafts in Sec. 33, par. 1.

The thickness of the crank webs on built crank shafts is not to be less than two-thirds of the diameter of the crank shaft, and the webs of solid crank shafts are to be in accordance with the following formula :—

$$b \times t^2 = \frac{d^3}{2 \cdot 5}$$

where d = Diameter of crank shaft,
b = Breadth of crank web,
t = Thickness of crank web.

* Applies to cases where the cranks are so arranged that two impulses occur simultaneously.

The radius of the fillets is not to be less than ·07*d.* for main bearings and ·05*d.* for crank pins.

(5) Cylinders must be water-cooled and the water jackets provided with test and drain cocks. An escape valve should be fitted on or near each cylinder head, in order to give warning should the pressure in the cylinder become excessive.

(6) Main engine cylinder heads, cylinders not fitted with liners, and cylinders for air compressors for starting and fuel injection purposes, are to be tested by hydraulic pressure to twice their working pressure. Cylinder water jackets are to be tested to 50 lbs. per square inch.

(7) The engine bed plate is to be well ribbed and strongly constructed, and the columns are to be carefully designed to withstand the tension stresses. When the engines are of the enclosed type, portable doors are to be fitted for the inspection of the cranks, bearings, etc., the crank cases to be provided with efficient means of ventilation. The engine-room platforms are to be supported on metal framework, not on wood.

(8) Compressed air reservoirs are to have sufficient capacity to ensure ample manœuvring power for the main engines. The material for cylindrical Air Holders constructed of riveted steel plates is to be of the quality and subject to the tests specified in Section 32 of the Rules for Boiler Shell Plates. The thickness of the plates is to be governed by the formulæ for cylindrical boiler shells.

Plans showing details of riveting, etc., are to be submitted for approval.

Seamless and Welded Holders are to be made from steel of the best Mild Open Hearth Quality having a tensile strength of not less than 23 tons per square inch, and showing an elongation of at least 25 per cent. on a gauge length of 8 inches.

The thickness of material in seamless and welded holders is to be in accordance with the following formulæ :—

$$\text{Cylindrical shells, } T = \frac{W \times D}{C} + \tfrac{1}{16}$$

$$\text{Flat ends, } \quad t = \sqrt{\frac{W \times D^2}{75000}} + \tfrac{1}{8}$$

$$\text{Dished ends, } \quad t = \frac{W \times R}{10000} + \tfrac{1}{8}$$

T = Thickness of shell, in inches.
t = Thickness of ends, in inches (to be in no case less than T).
W = Working pressure, in lbs. per square inch.
D = Greatest internal diameter, in inches.
R = Internal radius of dished end, in inches.
C = 15,500 for welded cylinders.
C = 22,000 for seamless cylinders with a tensile strength of 23 tons per square inch; with higher strengths the value of C may be proportionately increased.

All material is to be inspected and tested by the Surveyors to the British Corporation Registry.

The holders are to be submitted for inspection before the ends are closed.

Longitudinal seams are in all cases to be lap-welded. Electric, Oxy-Acetylene or Oxy-Hydrogen welding is not to be used for any part of the holders.

Seamless and welded holders are to be annealed after completion, and all holders are to be subjected to a hydraulic pressure test of double the working pressure, in presence of the Surveyor.

Openings for internal inspection are to be provided in the ends of the holders. When the holders are over 6 feet in length there should be openings in both ends.

A manhole is to be provided in all holders which have sufficient diameter.

All air holders are to be fitted with drain valves to draw off any accumulation of water or oil, and an internal pipe is to be fitted to the air inlet of the injection air holder, extending about half-way down the holder.

The air system is to be provided with a relief valve so constructed that it cannot be overloaded, and it is recommended that a non-return valve or other effective means be provided to prevent an ignition in the fuel valve casing from travelling through the injection air pipe to the air reservoir.

(9) Air compressors should not draw air from the crank cases of the engines. The air is to be efficiently cooled after each compression stage, and should have a final temperature not exceeding 20 deg. Fahr. above that of the cooling water before being allowed to enter the compressed air reservoirs. The compressors are to be fitted with safety valves, pressure gauges, and satisfactory arrangements for preventing the entrance of dirt and the delivery of oily air. The cooler coils are to be readily accessible for cleaning and removal.

Auxiliary compressors are to be fitted for use when the main engines are being manœuvred, and when the air compressor on the main engines is not available; they should not be of less power than half that of the compressors for one set of main engines. Arrangements should be made to provide water circulation from an independent pump in the event of a breakdown in the circulating pump.

(10) When vessels are intended to navigate shallow waters, two inlet valves for water circulation should be fitted, one on the ship's bottom and the other above the turn of the bilge. There should be in all cases an efficient strainer between the inlet valves and the circulating pumps, so designed and arranged that it may be cleaned and overhauled while the engines are working. Where there is no sight discharge from the cylinder jackets, a test cock or other means must be provided to enable the Engineer to satisfy himself that there is a continuous flow of water through each jacket.

(11) The fuel oil tanks are to be of sufficient strength to withstand the stresses due to the tanks being partly full when at sea; they are to be arranged so that leakage or drips will drain into wells having separate pumps; satisfactory ventilating arrangements are to be provided; and diaphragms of strong meshed wire gauze are to be fitted to all air pipes. The flash point of fuel oil carried in uninsulated tanks is not to be below 150° F., and such tanks are to be tested with a head of water at least 18 feet higher than that to which they are subject in practice. Where oil of a lower flash point is used, the tanks and connections are to be tested to at least 15 lbs. per square inch, and the tanks in sea-going vessels are to be separated from the engine room and cargo spaces by coffer dams. Where the fuel oil is carried in the

double bottom, an ample reserve is to be carried in a tank fitted separate from the double bottom, easily accessible for cleaning, etc., and fitted with internal coils for warming the oil when necessary. Fuel tanks which are worked under pressure are to be fitted with safety valves loaded to 5 per cent. above the working pressure, which discharge overboard or to the atmosphere above deck through wire gauze diaphragms.

(12) Filters for the fuel oil are to have bolted covers. Escape valves are to discharge into pipes leading back to the tank or to the atmosphere above deck; in the latter case the upper ends of the pipes are to be turned down and fitted with wire gauze diaphragms.

(13) All fuel and compressed air pipes are to be made of steel or annealed seamless copper, metal to metal joints are recommended where the pressure exceeds 400 lbs. per square inch. Exhaust pipes which pass through wood decks or close to combustible material must be effectively insulated. If the exhaust pipe is led overboard near the water line, it must be so arranged that water will not syphon back to the engine.

(14) All silencers are to be constructed so that they can be readily opened up for cleaning and inspection.

(15) Oil cooling tanks for forced lubrication are to be fitted with cooling coils connected to the water circulating system. The oil pumps are to be fitted with pressure gauges, and should be so designed that they cannot become air-locked. The oil well is to be so arranged as to minimise the risk of drawing air when the vessel is in a sea way.

(16) All electric ignition leads must be well insulated and protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may come in contact with oil. Commutators must be enclosed, and sparking coils must not be placed where they are exposed to explosive vapours. Exposed spark gaps are not to be fitted. Where lamps are used for ignition or for vapourising the fuel for paraffin and heavy oil engines, they should be fixed on a suitable bracket and the flame enclosed when in use.

(17) Spare gear in accordance with the following list is to be supplied, and stowed where readily accessible:—

2 Main bearing bolts and nuts.
2 Connecting-rod bottom end bolts and nuts.
2 Connecting-rod top end bolts and nuts.
1 Set bolts or studs for cylinder covers.
Packing rings for pistons and trunks of main engines and of each auxiliary internal combustion engine.
Piston rings for each stage of main and auxiliary air compressors, and for scavenging air pumps.
Fuel and air admission valves and exhaust valves, with seats and springs, for main engines and for each auxiliary internal combustion engine.
Suction and delivery valves and seats for main and auxiliary air compressors, scavenging air pumps, oil fuel, water circulating, and lubricating oil pumps.
Springs of each size.
Assorted bolts, nuts, studs, bar and plate iron.

Periodical Surveys.

Internal Combustion Machinery is to be subject to annual survey, at which a general examination is to be made of the main engines and their auxiliaries, such as may be necessary for the Surveyor to satisfy himself as to their efficient state. At least one cylinder of each engine, all the air compressors and coolers, the oil filters, the scavenging air and water-circulating pump valves, and at least half the number of air reservoirs are to be opened up and examined. The electric ignition, if any, and all the safety valves are to be examined and, as far as practicable, tested.

Special Periodical Surveys are to be carried out at the same time as the Special Surveys on the Hull, when, in addition to the requirements for Annual Survey, the following parts of the machinery are to be opened up and examined:—Cylinders, cylinder covers, needle valves and all other fittings on same, pistons, crossheads, thrust block, main and intermediate shaft bearings, shafting and steering gear. All oil tanks and air reservoirs are to be carefully examined and tested under water pressure, as required by the construction rules, and such other parts are to be examined as may be considered necessary.

The arrangements for pumping from the vessel's holds, as well as from the machinery spaces, are to be inspected; and while the vessel is in dry dock, all openings to the sea, together with the cocks and valves in connection with the same, examined. In addition, all iron and steel fastenings of sea cocks and valves to the shell plating should be removed for examination at each No. 3 Special Survey.

The propeller shaft should be drawn once every two years, and more frequently if considered necessary by the Surveyor; but when liners are fitted solid in one length, the shaft need only be drawn once every three years. The Committee are, however, prepared to consider representations from Owners as to special circumstances, which may modify these requirements in particular cases. When the after bearing is worn down $\frac{1}{4}$ inch with shafts not exceeding 9 inches in diameter, $\frac{5}{16}$ inch when over 9 inches and not exceeding 12 inches, and $\frac{3}{8}$ inch with shafts over 12 inches in diameter, the bearings must be rebushed.

CHAPTER XVI.

TUNING-UP, TEST BENCH AND ACCEPTANCE TRIALS OF DIESEL ENGINES.

To start up a Diesel engine, the following operations must be carried out:—

The crank of the cylinder provided with the starting air valve is turned to the starting point—*i.e.*, 20° past the top dead centre in the direction of rotation—so that the nose of the starting air cam is under its lever and the valve is open.

The cooling-water system is put into action, the cock leading to the fuel oil reservoir is opened, and the lubricators of the engine are adjusted.

The regulating lever of the fuel injection pump is brought to the starting position, and the valves of the fuel injection air reservoir are opened to put them into communication with the air compressor and with the cylinder head fuel injection valves.

Finally, the starting air reservoir is put into communication with the cylinder starting air valves. The engine commences to turn under compressed air, and when a sufficiently high speed of revolution has been attained, the valve gear control lever is brought to the running position, and the engine is then put on fuel.

If the engine does not pick up on fuel, this lever is brought back to the starting position, in order again to increase the revolutions before attempting for the second time to get the engine to fire on fuel. In the event of the engine failing for the second time to pick up on fuel, it may be assumed that there is some defect in the mechanism, or some mistake in erecting, and it is preferable to seek for this rather than to waste the air storage in useless attempts to start.

The pressure of the starting air should preferably not be greater than from 30 to 45 atmospheres (425 to 640 lbs. per square inch), as it may happen that leaky starting air valves may cause the compression to increase to such an extent that the engine is stopped and swings in the reverse direction of rotation, whilst it is exceedingly dangerous to subject the engine to excessively high pressures.

At starting, the fuel injection air should have a pressure of about 50 atmospheres (710 lbs. per square inch), as at light loads higher injection pressures cause faulty running.

When the engine has been started and the starting air reservoirs * re-

* If, due to any mistake or to a series of failures to start, it should occur that all the reservoirs have been emptied, they may be recharged from a bottle of CO_2, and, to accelerate this operation, the pipe communicating between the CO_2 bottle and the engine reservoirs may be heated by means of the application of waste soaked in hot water. The CO_2 bottle itself must never in any way be directly heated, or accidents, as has been the case, may result.

charged, the delivery of the compressor is regulated as described in Chapter XIII., and the attention of the man in charge may be limited to careful supervision of the lubricators, the pressure in the reservoirs, temperature of the cooling water, etc.

It is difficult to give more exact instructions for the running of Diesel engines, nor would such fall within the scope of this work.

Each design of engine contains peculiarities which must be taken into consideration, and concerning which the constructors of the engines always supply information to those to whom the running of the engine will be entrusted. In addition, it can be stated that every engine has its own little running peculiarities, which can only be learned by the engineer in charge as a result of considerable experience.

The frequency with which inspection and overhaul of the various parts should be carried out depends largely on local conditions, such as the quality of the fuel and lubricating oils, the kind of load to which the engine is subject —*i.e.*, whether light or heavy, variable or constant.*

Only experience can teach the periods of time after which it is desirable to clean or grind in the main exhaust and suction and fuel injection valves, the air compressor and the fuel injection pump valves.

The various adjustments and trials which are periodically carried out are similar to those which are required to be made when the engine is new.

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*The following table, taken from the instruction-book issued by Messrs. Sulzer to those in charge of engines supplied by them, may serve to give a brief idea of the order and frequency with which inspection and overhaul should be carried out for the usual type of four-cycle stationary engine running under normal conditions:—

	One-cylinder Engine.	Two-cylinder Engine.		Three-cylinder Engine.		
		1st Cylinder.	2nd Cylinder.	1st Cylinder.	2nd Cylinder.	3rd Cylinder.
1st Week.	Air compressor valves. Filters of fuel tank. Lubrication system to main bearings.	Air compressor valves. Lubrication system to main bearings.	Exhaust valve.	Air compressor valves. Fuel injection pump valves. Lubrication system to main bearings.	Exhaust valve.	Air compressor valves. Filters of fuel tank.
2nd Week.	Exhaust valve.	Exhaust valve. Fuel injection pump valves. Filters of fuel tank.	Air compressor valves.	Exhaust valve.	Air compressor valves.	Exhaust and starting air valves. Suction valve. Fuel injection valve.
3rd Week.	Air compressor valves. Fuel injection pump valves.	Air compressor valves.	Exhaust and starting air valves. Suction valve. Fuel injection valve.	Air compressor valves.	Exhaust and starting air valves. Suction valve. Fuel injection valve.	Air compressor valves.
4th Week.	Exhaust and starting air valves. Suction valve. Fuel injection valve.	Exhaust and starting air valves. Suction valve. Fuel injection valve.	Air compressor valves.	Exhaust and starting air valves. Suction valve. Fuel injection valve.	Air compressor valves. Lubricating oil filter.	Exhaust valve.

After One Month.—Examine the pistons of the air compressor.

After One Year.—Examine the main pistons, and renew the piston rings as required; clean the main fuel oil tank and the inside of exhaust silencer.

Test Bench Trials.—The most important trials to be carried out with the engine are those made before the engine leaves the makers' works, described as tuning-up trials, and those when the engine has been erected in place—*i.e.*, acceptance trials.

The bench trial requires great experience and not a little prudence on the part of those responsible, since explosions, which are part of the principle upon which the internal combustion works, may be so accentuated by any small defect in construction or by carelessness in erection as to produce disastrous effects upon the engine and serious consequences to those in charge. For this reason it is necessary, before the first start is attempted, after the engine has been erected on the test bench or erected in place, to make quite sure that no mistakes have occurred in the erection. The fuel injection pump should be tested to see that all joints are tight and that no air is present. The clearance between the valve lever rollers and their respective cams should be adjusted. The various valves should be opened by hand, to ensure that they return readily to their seat under the action of their springs, and finally, the engine should be turned by means of the turning gear for a few revolutions to make sure that all is clear.

Whilst the engine is being turned by hand, the cylinders should be in communication with the atmosphere by means of an arrangement of the type shown in Fig. 220, p. 154, or preferably, by opening the cylinder indicator cocks, so that it may be seen whether, due to any defect, any water has entered the cylinder, which, if it were allowed to remain, might cause serious consequences.*

During the early trials of a new engine, lubricating oil should be supplied in excess, as however good the workmanship, the bearing surfaces are not as smooth, nor is the contact so good, as when the engine has run for a certain period. The friction and the chances of bearings seizing are far greater during the early periods of running than when the engine has been for some time in regular service.

The compressed air leads from the air reservoirs to the fuel injection valves on the cylinders, and those from the air compressor and the starting air pipes should be tested by air under pressure whilst the engine starting lever is in its neutral or stop position (Position II, Fig. 200, p. 142). Large leaks make themselves heard at once, and smaller ones may be discovered by pouring a little oil over any suspected point, when any leak makes itself apparent by the formation of bubbles.

If the air compressor receiver is provided with a pressure gauge, and the shut-off valve on the pipe leading from the air compressor to the air reservoirs be opened, the H.P. delivery valve is tested for tightness, since if the gauge should rise, it may be inferred that the H.P. delivery valve is not perfectly tight and is passing a certain quantity of air.

When an engine has been completely erected, the foregoing tests should be carried out, and are generally sufficient if the engine has previously been run, but in the case of a new engine, the following tuning-up trials must in addition be carried out.

* The smallest quantity of water in the cylinder diminishes the available compression volume for the air at the end of the compression stroke, and may be the cause of exceedingly high cylinder pressures, giving rise to dangerous stresses in the cylinder head and the piston.

The condition of an engine about to be started for the first time requires careful inspection to ensure that it is provided with all the necessary mechanism for its work; even then it may not be definitely known whether the simultaneous working of these various parts will be so harmonious as to make it run.

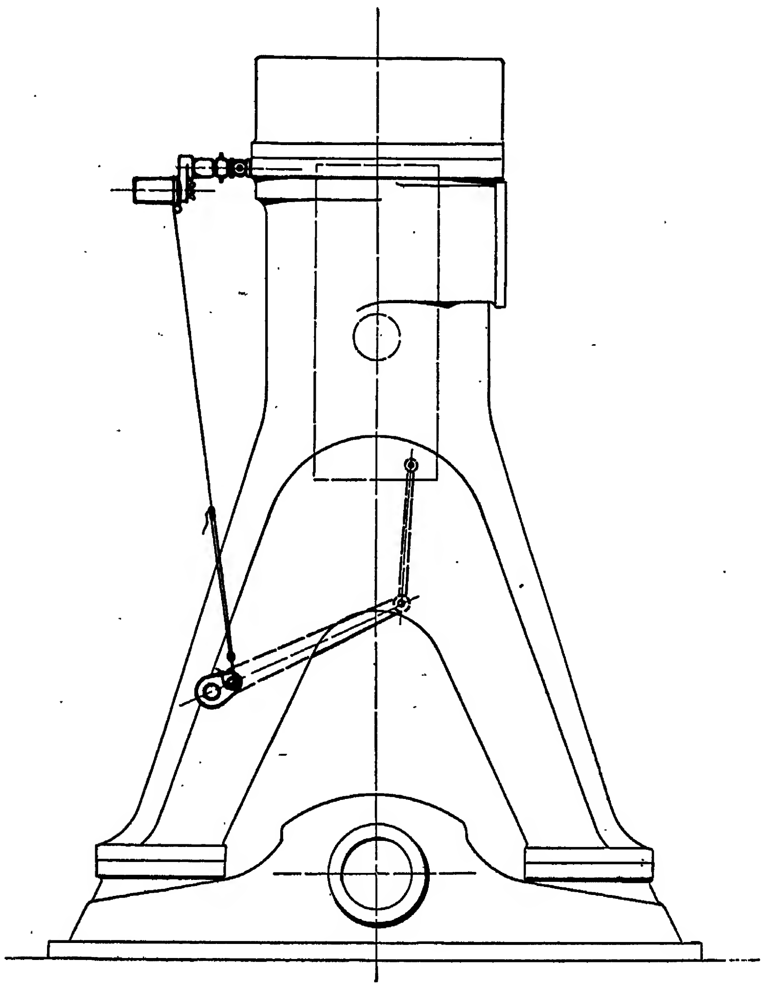

Fig. 368.—Indicator Gear worked from the Main Piston.

During the first few revolutions, those responsible for running the engine should pay the strictest attention to the working parts; it is quite unnecessary to be concerned as to whether ignition is taking place regularly or as to the condition of the exhaust, since during the first few revolutions the probability of the main bearings heating up and the pistons seizing is

greatest. The governor, which may be stiff and irregular, should be carefully watched to ascertain that it does not permit of the speed of revolution exceeding the maximum; in fact, it is preferable to run the engine for a short time at a slow speed.

After a few minutes' running, it is generally necessary to stop the engine to remedy those defects which are already apparent, after which the engine may be started again and allowed to run light for an hour or so until all the bearings are run in.

After this it may be gradually loaded by any of the ordinary systems of brakes suitable for the power desired,* and the condition of the pistons,

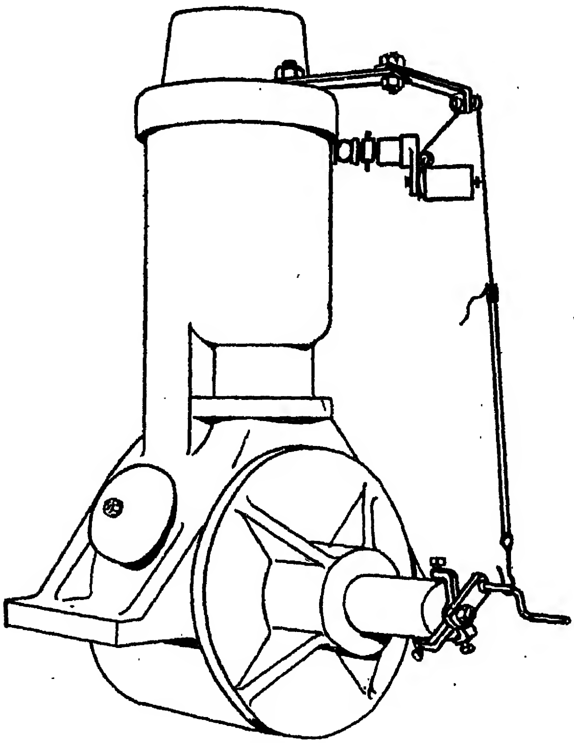

Fig. 369.—Indicator Gear Crank Clamped to Crank Shaft.

which heat up with increased load, should be ascertained by applying the ear to the engine framing; if dull knocks are heard, the engine should be

* A dynamo is exceptionally suitable for use as the brake against which the Diesel engine is to be run, provided that its efficiency is known and that the switchboard instruments are reliable.

The Prony brake is the simplest and most economical power-measuring apparatus, and is well adapted for powers up to 100 B.H.P., although, in the hands of experts, 300 B.H.P. may be absorbed by this type of apparatus. For high powers the Froude dynamometer is most widely adopted.

immediately stopped and the piston withdrawn for examination to find out whether seizing has commenced. With Diesel engines it is found that this knocking only precedes piston seizure by a few moments.*

When the engine is fully loaded the tuning-up trials are commenced and indicator diagrams are taken to ascertain the valve settings, etc. Almost all Diesel engines are provided with indicator gear which may be worked from the main piston (Fig. 368), or from special eccentrics fitted to the intermediate vertical shaft (Tosi), or on the cam shaft for two-cycle engines.

Many other types of arrangements may be adopted, which perhaps give simpler mechanism but less accurate diagrams, as, for example, a three-arm clamp carrying a small crank which may be fitted to the end of the crank shaft and adjusted by the eye to be in line with the crank of the cylinder from which it is desired to take the indicator diagrams (Fig. 369).

If a small tapped hole is provided in one end of the crank shaft, the indicator crank may be held as shown in Fig. 370.

With slow-running Diesel engines, the indicator diagrams may be worked out to obtain the indicated H.P., and the mechanical efficiency is known by comparing the figure so obtained with the B.H.P. as obtained at the brake; but with high-speed engines the vibrations of the indicator cord, especially

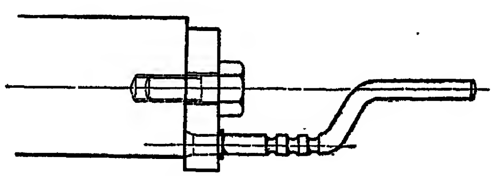

Fig. 370.—A Method of Fitting Indicator Gear Crank to Engine Crank Shaft.

if a long cord is used, and the influence of the momentum of the moving parts of the indicator, are such that the result obtained by working out the diagram is hardly reliable, and is often omitted.

In the tuning-up trials of both slow- and high-speed engines, indicator diagrams play a most important part, since they are the best means of revealing any defect in the engine, and if care be taken not to allow the pencil to draw the diagram of more than one complete cycle, an accurate interpretation is relatively simple.

Sometimes diagrams are irregular, due to defective indicators, and Fig. 371 gives an example of a diagram where the indicator cord is too long and the indicator drum comes against its stop before the end of the engine stroke, whereas in Fig. 372 a diagram is shown in which the cord is too short.

If the indicator piston is working stiffly, due to dirt or lack of lubrication,

* Cylinders and pistons, however well machined, are never truly cylindrical, and when, during or after shop trials, the engine is dismantled, the pistons can be filed with a large smooth file where the most polished surfaces show the greatest wear to have taken place.

jerky movement of the pencil is caused, and a diagram similar to that shown in Fig. 373 may be obtained.

For high-speed engines (500 to 600 revolutions per minute), unless special indicators be used, the curves are generally made up of wavy lines, similar to those obtained when the vibration of the indicator cord is excessive (Fig. 374).

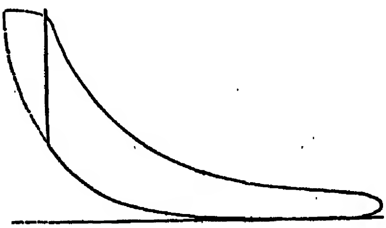

Fig. 371.

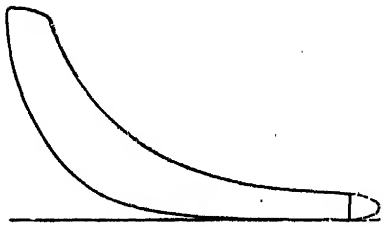

Fig. 372.

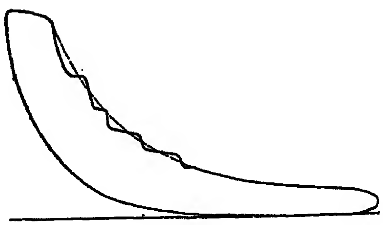

Fig. 373.

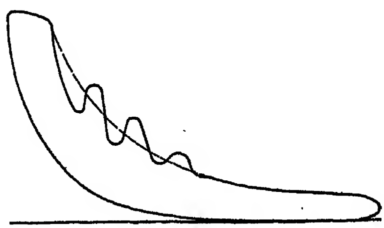

Fig. 374.

Figs. 371 to 374.—Diagrams illustrating Irregularities due to Defective Indicator or Gear.

The operations of tuning-up a Diesel oil engine may be classified as follows :—

(*a*) Checking and adjusting the compression pressure.
(*b*) Checking the timing of the valve gear.
(*c*) Adjustment of the fuel injection valve.
(*d*) Adjustment of the governor.
(*e*) Regulation of the fuel injection pump.

(*a*) The value of the final compression pressure is arrived at by taking an indicator diagram from the engine when it is hot and with the fuel injection pump and fuel injection air supply shut off, and whilst it is still running under the influence of the flywheel. The scale of the indicator spring being known, the highest point of the curve of the diagram will give the maximum pressure reached.

It might seem that in this diagram the curves of expansion and compression ought to be superimposed. In practice this never happens, owing to

the exchanges of heat which take place between the cylinder walls and the air in the cylinder.*

The final value of the compression pressure varies from 29 to 35 atmospheres (410 to 500 lbs. per square inch) according to the makers of the engine, and in two-cycle engines even reaches 36 atmospheres (510 lbs. per square inch).

In any case, if the pressure obtained from the diagram taken in the manner just described should, owing to some fault in erecting, be different from that desired,† it will be necessary to correct it by filing the feet of the frames to increase the pressure, or by taking a thin cut off the crown of the piston in the lathe if it is required to lower the pressure. When connecting-rods of the form shown in Figs. 116 and 117 (p. 98) are adopted, the same effects may be obtained with less trouble by fitting or removing thin liners between the rod and the bottom end brasses.

Contractors usually provide tables indicating by how many tenths of a millimetre it is necessary to alter the distance between the piston crown and the cylinder head in order to obtain a given variation of the maximum compression pressure.

All the same it is unusual, in an engine constructed by a good firm, to have to modify the compression during the trials, for with accurate workmanship the results aimed at are almost always obtained immediately.

(*b*) To check the valve settings, it is sufficient to turn the engine by hand and ascertain by means of a spirit level, adjustable to any angle, held against the crank arm, whether the opening and closing of the valves take place at the desired angle. The angle during which a valve is considered to be open is that through which the crank shaft turns from the moment the roller of the valve lever comes in contact with its cam to the moment when this contact ceases.

The indicator diagram also shows whether the valve settings, especially the entry of the fuel and the exhaust, are properly adjusted. Regarding the first, mention will be made when treating of the adjustment of the fuel

* Leaky valves or piston might also be the cause of the curves not being superimposed. A leaky piston may be easily detected, however, by the sound, which is like a sneeze, and is shown by the normal diagram of the engine assuming the form indicated in Fig. 375.

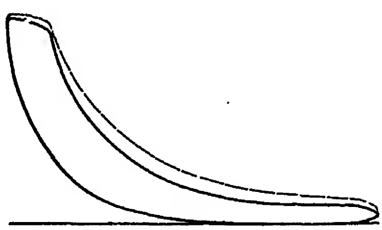

Fig. 375.—Diagram giving evidence of Leaky Valves or Piston.

† Dealing, as is the case, with very high pressures, a difference of a few thousandths of an inch in the distance between the piston crown at the dead centre and the cylinder head produces an appreciable difference in the value of the maximum compression pressure.

injection valve; with regard to the exhaust, if this commences too late the diagram will assume the shape shown in Fig. 377, whilst if it is too early the indicator card will be similar to Fig. 376.

(*c*) The adjustment of the fuel injection valve is the most difficult, delicate and important part of the regulation of a Diesel engine.

There are two functions which this valve has to fulfil; that of suitably timing the entry of the fuel into the cylinder, and that of insuring that the entry shall take place in such a way that combustion may occur practically at constant pressure.

If the opening of the fuel injection valve commences too soon the diagram

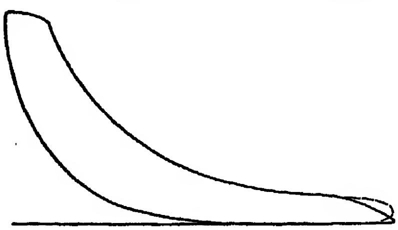

Fig. 376.

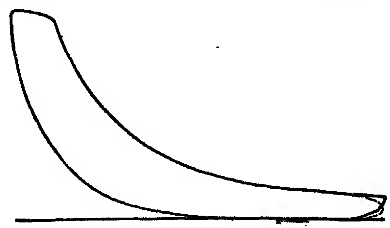

Fig. 377.

Figs. 376 and 377.—Diagrams illustrating Bad Timing of Exhaust.

assumes the appearance of that shown in Fig. 378, whilst if the opening is too late the diagram will have the form shown in Fig. 379.

The similarity of the two forms may allow some doubt as to the nature of the defect, but all uncertainty may be eliminated by comparing the diagram with that of the compression, obtained, as previously described, with the engine running under the momentum of the flywheel, and with the fuel injection valves shut off.

If the maximum pressure obtained from the diagram under examination

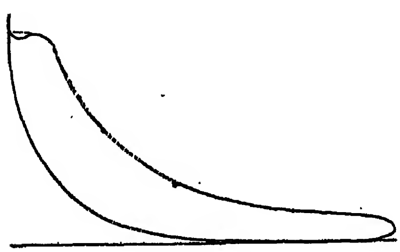

Fig. 378.

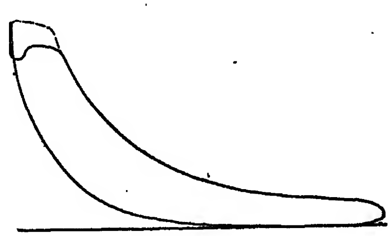

Fig. 379.

Figs. 378 and 379.—Diagrams illustrating Bad Timing of Fuel Injection.

is greater than that of the compression, the injection of the fuel is too early. If the maximum pressure is equal to that of compression and the diagram continues at a lower pressure the injection of the fuel is late.

Moreover, too early injection is almost always accompanied by knocking at the moment of ignition.

To correct either of these defects it is sufficient to move slightly the

point of the fuel injection valve cam, an operation rendered easy by its construction (Fig. 217, p. 157).

Small corrections may be obtained by adjusting the clearance between the cam and the roller of the lever by means of the adjusting screw, without, however, making the clearance so small as to keep the roller in continuous contact with the cam, or making it so great that the lift of the valve becomes too small or occurs with a jerk.

The diagrams for adjustment of the fuel injection valve should be taken whilst the engine is working at full load or overload, as in these conditions the diagram is always clearer and more regular. To determine the instant of the injection of the fuel it is necessary, moreover, to wait until the engine is well warmed up, since on starting the engine diagrams frequently show a slight retardation, which, after the engine has run for some time, may disappear, since the fuel then becomes more fluid and the temperature of the combustion chamber rises.

The factors influencing the adjustment of the fuel injection valve, as regards its function of introducing the fuel so that combustion shall take place at constant pressure, are the following:—The pressure of the injection air; the diameter of the hole in the steel washer O (Fig. 184, p. 132); the

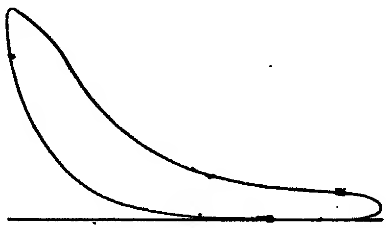

Fig. 380.—Diagram showing Falling off of Pressure during Injection owing to Low Injection Air Pressure.

number of the perforated rings; and the number and the diameters of the holes in them. It is only by trying several combinations of these factors, and with much patience and practice, that it is possible to obtain good results.

It is necessary, however, not to make the diameter of the holes too great nor the fuel injection air pressure too high, lest the consumption of compressed air should be excessive or the reserve output of the compressor reduced.

Even with diagrams of the form represented in Fig. 380, which often do not require a great expenditure of air, it is possible to obtain small consumptions of fuel and a smokeless exhaust.

The fuel injection air pressure required to obtain a good diagram varies with the load, but depends also upon the design and construction of the fuel injection valve. There are engines which may be run at overload with a fuel injection air pressure of 60 atmospheres (850 lbs. per square inch), whilst others require 75 or even 80 atmospheres (1,000 or 1,150 lbs. per square inch).

Frequently, with engines that have already been tuned-up, the presence of smoke in the exhaust may be attributed to a disproportion between the injection air pressure and the load. If the pressure is too low the smoke is

black, if too high, it is frequently whitish, and the ignition is accompanied by knocks—similar to those produced by premature ignition.*

High-speed engines require higher injection air pressures, and sometimes also a longer period of injection than do slow-running ones; with the former it is almost impossible to obtain a perfectly clear, colourless exhaust. This will be understood on considering how extremely short is the time during which the phase of combustion has to take place, a period which frequently does not reach 100th of a second.

With engines of the two-stroke cycle, it is also very difficult to obtain a smokeless exhaust, even if combustion inside the cylinder is good. The smoke in this case is to be attributed to lubricating oil escaping from the exhaust ports into the exhaust pipe, where it volatilises and decomposes on contact with the hot walls and with the high temperature gases of exhaust.

Not only are the quantity and the colour of the smoke most important symptoms of the manner in which combustion is taking place, but another indication is also furnished by the lower part of the piston which overruns the cylinder at the lower dead centre; if the combustion is good this part remains bright and clean; it becomes black and dirty with burnt oil if the combustion is bad.

(*d*) The first problem of the regulation of the governing mechanism is that of adjusting the revolutions to the number required. For this purpose it is sufficient to count the number of revolutions per minute † when the engine is under normal load, and to slacken or tighten the spring of the governor according to whether it is desired to reduce or increase the speed.

With the variation of load the speed should vary slightly; between running at full load and running light, the difference of speed should not be greater than about 4 per cent.

To judge of the quickness of action of the governor, it is sufficient to observe the variations of a tachometer driven by the engine crank shaft during a sudden alteration of load. With new engines the governor gear is always stiffer than with those which have been running for some time, on which account results may be accepted on the test bed, which during normal running would not be altogether satisfactory.

The oil brake (dashpot), which is always fitted to the governor, ought to be so adjusted as not to interfere with the quick action of the latter, whilst at the same time preventing it from oscillating continuously, or hunting.

Some engines, especially those for central power stations and high-power marine engines, are provided with a safety governor, which operates when the speed exceeds a certain number of revolutions, stopping the engine at once. The adjustment of this governor is also quite simple, the tension of its spring being increased or decreased until it is found to operate at the desired speed of revolution.

(*e*) The fuel injection pump is always provided with mechanism which permits variation of the delivery to correspond to any given position of the

* On p. 226 is described the apparatus employed with high-power two-cycle engines for automatically varying the fuel injection air pressure according to the variation of the load.

† When counting the revolutions it should be remembered to start at "nothing"; by commencing with "one," the result for the minute would be one greater than the real number of revolutions.

governor. For pumps of the type shown in Fig. 274 (p. 197) this mechanism is as described on p. 200 (Fig. 277), whilst for those of other types it may be reduced to a means for lengthening or shortening the link connecting the governor to the distributing mechanism of the pump.

In adjusting the fuel injection pump, the delivery is at first arranged to be excessive, and then reduced little by little, keeping the engine at overload until the collar of the governor descends almost to its lowest position. It is evident that at lower powers the governor collar will be raised higher and the delivery of the pump will diminish as described in Chapter XII.

With multi-cylinder engines, each cylinder should develop the same power. This condition is very important, since if it is not satisfied it may happen when running at high power that one of the cylinders, doing more than its share of the work, may be overloaded, and consequently be damaged.

In order that the different cylinders should work at equal powers, they must be supplied with equal quantities of fuel. For this reason, with engines having a single fuel injection pump and a diaphragm distributor, it is necessary to calibrate the holes of these diaphragms as mentioned on p. 206, so as to compensate for the different losses of pressure caused by the different lengths of piping leading to the various fuel injection valves; with engines having an independent pump for each cylinder it is necessary that the pumps should all have the same delivery.

In both cases the adjustment may be made thus:—The engine is run at a light load with only one cylinder firing, and the position assumed by the governor in this condition is marked. Then this cylinder is shut off and another started up, and its power so adjusted that the governor assumes the position which it had when the first cylinder was working. When this is done in the case of all the cylinders of the engine, the power will be the same in each. Then starting up all the cylinders together, and overloading the engine, it will be seen whether the total delivery of the fuel injection valves is correct, controlling the position of the governor as described for single-cylinder engines.

Bench Trials.—The indispensable trials on the test bench for a Diesel engine are:—

(*a*) The trial at normal power.
(*b*) The trial at maximum guaranteed power.
(*c*) The fuel consumption trial.

If it is wished to make more exhaustive trials, the following may also be determined:—

(*d*) The mechanical efficiency—separating the real mechanical efficiency from the work absorbed by the compressor.
(*e*) The distribution of the heat losses.

(*a* and *b*) With engines of not very great power and almost always in the case of those for industrial purposes, by the normal and maximum guaranteed powers are intended brake horse-power.

Only for very large marine engines is the contract sometimes made for the indicated horse-power.

The number of B.H.P. given by the engine is determined by one or other of the usual methods of braking, provided it is suitable to the power.

To ensure that a trial at normal power may give reliable results it should be of long duration.

Time is necessary that the heat may spread throughout the engine, and that the various parts may reach the temperatures they will have under conditions of continuous running. It must not be assumed that an engine which develops a given power for two or ten minutes can do so for two hours, and still less for five or six.

Since it is not always possible to find a plausible technical explanation for the fact that some part has worked satisfactorily, perhaps for a whole day, or even for a month, and breaks or gives out after another hour's running, there is the temptation to say that the engine becomes *tired*.

Hence, for an important plant and for a high-powered engine, the trial cannot ever be too long, and it is not unreasonable for the purchaser of an engine to stipulate for a continuous trial of 24 hours or more.

For the same reasons, on the other hand, if the maximum power is guaranteed as *temporary*, it is not fair to insist on the engine developing it for too long a period. If after running for an hour or two at overload a piston should seize or a bearing become overheated, it would be the fault of the person responsible for the engine being run in this way, even if during the first ten minutes the engine developed the power with the greatest ease.

During the trials the revolutions should be noted regularly and diagrams taken periodically.

If, having regard to the speed of the engine and the reliability of the indicator adopted, it is thought that the results of working out the diagrams may be trustworthy, it is as well to calculate the indicated horse-power and the *gross* mechanical efficiency.*

(*c*) The fuel consumption trial ought to last at least one hour, better still two, for if a shorter time is taken the inaccuracies of measurements have too great an influence on the correctness of the results.

The usual procedure is as follows :—At the commencement of the trial the fuel reservoir is filled to a certain level, which is accurately marked, and at the end of the trial the fuel necessary to bring the level in the reservoir to the original mark is weighed.

Then the total weight of fuel used is divided by the B.H.P. developed, and by the duration of the trials in hours, to obtain the consumption per B.H.P.-hour.

(*d*) To obtain the *net* mechanical efficiency, it is only necessary to take into account the indicated H.P. absorbed by the compressor.

$$\eta_n = \frac{\text{B.H.P.}}{\text{I.H.P.} - \text{I.H.P. of compressor}}.$$

The required power of the compressor is obtained by integrating the diagrams of all its stages.

For a two-stage compressor having the diameter of the H.P. piston $= d$, and that of the L.P. $=$ D, and the common stroke S.

$$N_{ic} = \frac{\frac{\pi}{4} d^2 \times p_m \times S \times n}{75 \times 60} + \frac{\frac{\pi}{4} D^2 \times P_m \times S \times n}{75 \times 60}.$$

$$N_{ic} = 1{\cdot}743\, S \times n\, (p_m \times d^2 + P_m \times D^2).$$

* By *gross* mechanical efficiency is meant that including the power absorbed by compressor or compressors. The *net* efficiency, on the other hand, is that when the work of the compressors is taken into consideration separately.

In this last formula the pressures are in kgs. per sq. cm. and the dimensions in metres.

(*e*) The thermal efficiency of the engine is obtained as stated on p. 53 from the formula

$$\eta_t = 635 \frac{N}{P \times h}.$$

The heat losses are divided into—heat absorbed by friction, radiation, imperfect combustion, and by cooling water, and contained in exhaust gases.

The heat absorbed by friction can be evaluated when the net mechanical efficiency is known. It is given in calories by—

$$K_a = \frac{N_i + N_{io} - N_e}{N_e} \times 635,$$

$$K_a \text{ per cent.} = \frac{K_a}{P \times h}.$$

The heat absorbed by the cooling water may be ascertained by measuring the quantity of water used per hour either by means of a meter or by collecting the water discharged from the jackets in a measuring tank, and taking its temperatures t, when leaving, and t_o, before entering the engine, then :—

$$K_h = G\,(t - t_o).$$

$$K_h \text{ per cent.} = \frac{K_h}{P \times h}.$$

Where G is the cooling wat

To calculate the heat los

be taken and analysed to measu

the carbon and hydrogen contained in the fuel must be ascertained and the temperatures of the suction air and the exhaust gas taken.*

The calories lost per kg. of fuel are obtained from :—

$$K_s = \left(0{\cdot}32 \frac{C}{0{\cdot}536\,k} + 0{\cdot}48 \frac{9\,H}{100}\right)(T - t),$$

in which—

C = the percentage of carbon contained in the fuel.
H = the percentage of hydrogen contained in the fuel.
k = the percentage of CO_2 in the exhaust gas.
T = the temperature, C°, of the exhaust gas taken close to the exhaust valve.
t = the temperature, C°, of the suction air.

The percentage of heat lost in the exhaust is given by :—

$$K_s \text{ per cent.} = \frac{K_s}{P \times h}.$$

The heat losses through radiation, imperfect combustion, etc., are taken together, and are generally considered to be the difference between 100 and the sum of the other percentages calculated.

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* Franz Seufert, *Versuche an Dampfmaschinen, Dampfkesseln, Dampfturbinen, und Dieselmaschinen*, 3rd edition, Berlin, Springer, 1913.

CONVERSION TABLE FOR BRITISH AND METRICAL MEASURES.

(Closely Approximate.)

Metric to British.	British to Metric.
Length— 1 m. = 3·28 feet = 39·37 inches. 1 cm. = 0·3937 inch = $\frac{1}{2\cdot54}$ inch. 1 mm. = 0·03937 inch = $\frac{1}{25\cdot4}$ inch.	1 inch = 25·4 mm.
Area— 1 m.^{2} = 1,550 sq. in. = 10·76 sq. ft. 1 cm.^{2} = 0·155 sq. in. 1 mm.^{2} = 0·00155 sq. in.	1 sq. inch = 6·45 cm.^{2}
Volume— 1 litre = 1,000 c.c. = 0·03531 cub. ft. = 0·22 gal.	1 cub. ft. = 28·316 litres = 6·229 gals.
Weight— 1 grm. = 0·002205 lb. 1 kg. = 1,000 grms. = 2·205 lbs. = ·0009842 ton.	1 lb. = 454 grms.
Pressure— 1 atmosphere, 1 kg. per cm.^{2} } = 14·22 lbs. per sq. inch. 1 kg. per mm.^{2} = 1,422 lbs. per sq. inch. = 0·634 ton per sq. inch.	1 lb. per sq. in. = 0·07031 atmos. = 0·07031 kg. per cm.^{2}
Temperature— 1° C. = $\frac{5}{9}$(° F. − 32°).	1° F. = $\frac{9}{5}$ °C. + 32°.
Energy— 1 kg. m. = 7·233 ft.-lbs.	1 ft.-lb. = 0·1382 kg. m.
Power— 4,562 kg. m. per min. = 33,000 ft.-lbs. per min. (one H.P.) 76 kg. m. per sec. = 550 ft.-lbs. per sec.	1 British H.P. = 1·01385 Metric H.P.
Heat— 1 calorie = 3·97 B.Th.U.	1 B.Th.U. = 0·252 calorie.
Power and Heat— One H.P. = 42·3 B.Th.U. per. min. = 10·68 cals. per min. One H.P. per hour = 2,540 B.Th.U. per hour = 641 cals. per hour.	
Electrical Units— 1 E.H.P. = 746 watts = 0·746 kw. 1 kw. = 1·5 B.H.P. (roughly.)	

INDEX.

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